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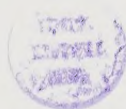
Heat transfer through windows

**During the hours of darkness with
the effect of infiltration ignored**

Bertil Jonsson

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**Swedish Council for
Building Research**



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HEAT TRANSFER THROUGH WINDOWS

During the hours of darkness with
the effect of infiltration ignored

Bertil Jonsson

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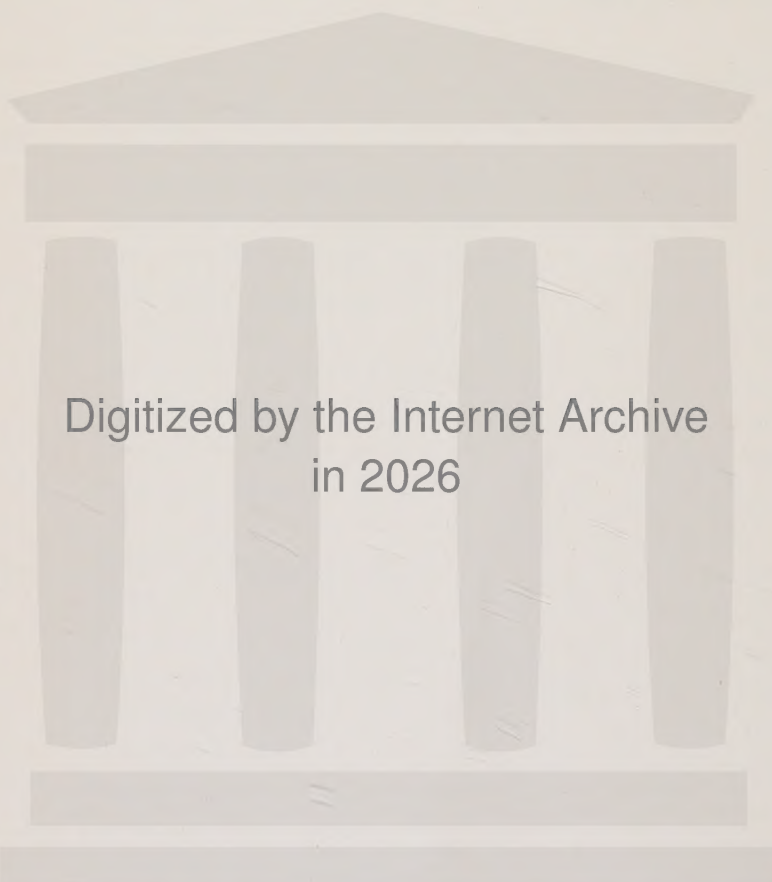
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FOREWORD

This report relates to the research project Heat Transfer Through Windows. The object of this project is to obtain by theoretical and experimental studies wider knowledge and practical experience of the complex interaction between conduction, convection and radiation which characterizes heat transfer through a window. A large proportion of the work has been devoted to studies of the convection process. Calculations have also been carried out to study the way in which heat transfer through the junctions between glass, casement and frame occurs for different methods of construction.

Professor Bo Adamson initiated the project and in this way opened the window for me to this area of research. During the course of the project he has made many valuable suggestions and observations. Bengt Eftring has given assistance with the computer program, and Urban Lundh, Rolf Månsson and Bernhard Murd with the experimental measurements. Johan Claesson and Gunnar Anderlind have checked the manuscript. Marianne Abrahamsson has typed the text and Lewis Gruber, Cambridge, has translated it into English.

To all the above and all the others who have directly or indirectly helped me in my work I wish to express my sincere thanks.

I also wish to express my thanks to Gullfiber AB, Bjuv, who have greatly assisted me by calibrating different test materials, and to Pilkington AB, Halmstad, who kindly supplied sealed units of different configurations.

Finance for the project has been provided by the Swedish Council for Building Research.

Lund, March 1985

Bertil Jonsson

SUMMARY

Transfer of heat through windows

Transfer of heat through an air space comprises an interaction between the three fundamental heat transfer mechanisms, conduction, convection and radiation. In a glass assembly and in a window different materials are also present, and a complex three dimensional flow therefore occurs.

All bodies, whether solids, liquids or gases, emit energy in the form of radiation and are also capable of absorbing such energy. This radiation which originates in warm bodies is usually referred to as infrared radiation or temperature radiation and forms part of the heat transfer process to a greater or lesser extent. In the case of windows it is particularly two wavelength regions which are of interest, namely infrared radiation at 5000-6000 K, i.e. the solar spectrum at the earth's surface ($0.3-2.0\mu\text{m}$), and long wave infrared radiation at about 293 K comprising wavelengths within the $3.5\text{--}20\mu\text{m}$ region. In the first of these regions, transmission characteristics can be varied to control insolation. For the long wave region a reduction in radiation causes heat losses to diminish. An energy economic window should admit the maximum insolation (100% transmission for $0.3 < \lambda < 2.0\mu\text{m}$) and have the minimum heat loss (nil transmission for $3.5 < \lambda < 20\mu\text{m}$). By applying layers which have different properties in different wavelength regions (selective layers), either directly to the pane of glass or in the form of transparent films, heat flow can be limited.

In an air space, transfer of heat between the solid surfaces and the intermediate medium takes place by a combination of conduction and convection. The quantity of energy supplied raises the internal energy of the fluid and is carried away by its motion. When the particles reach a region of lower temperature, heat is transferred by conduction to the solid surface. Energy transfer comprises an interaction between convection and conduction. The two mechanisms cannot be distinguished.

Calculation of natural convection in vertical spaces

For natural convection in vertical spaces, a mathematical model must be constructed which describes the fundamental equations. These are the continuity, momentum and energy equations. In general there are no analytical solutions to this system of equations, and a finite difference procedure must be resorted to. Two computer programs, one by G de Vahl Davis (1967) and another by R W Thomas (1974) have received particular study. These two programs have also been documented by B Efring (1974), (1975). It has been decided to make use of R W Thomas: Finite Difference Computation of Heat Transfer by Natural Convection. Dissertation of New South Wales (1970), for computation of the convection process. The computer program which refers to non-steady natural convection is valid for fluids enclosed between two vertical planes and two planes perpendicular to the vertical axis. Most computer programs which have been constructed for natural convection in a space relate only to the space as such and take no account of different conditions at the surfaces. The boundary conditions in these calculations are based on a uniform temperature of T_H and T_C on the warm and cold side respectively. Along the horizontal edges the temperature may be assumed to have a linear distribution between the cold and warm sides, or the edges may be assumed to be completely insulated from the space.

The problem is characterized by three nondimensional parameters, the Grashof number Gr , the Prandtl number Pr and the height/width ratio. The equations are expressed by finite differences. The space is covered by a rectangular mesh parallel to the axes of coordinates. In choosing the nondimensional time increments Δt and the number of nodes, a compromise must be made between small values (minimum errors, long computation times) and large values (short computation times but problems of instability). For high values of the Rayleigh number or large height/width ratios, instability problems then occur. The mesh must be refined and the time increments made smaller.

Runs have been made for values of the Rayleigh number $Ra = GrPr$ between 1000 and 400000. The height/width ratio has been varied between 1 and 120. Calculations have been made for different

gases in the space. It has been possible in these calculations to study the way in which the convective air current changes character when a variable is altered. Flow has been characterized into three flow regimes: the conduction, transition and boundary layer regimes. Velocity profiles have been computed for typical convection cases.

The local Nusselt number (Nu) has been calculated for these cases. This has been studied along the inner warm surface for varying vertical positions. In this way the average value could also be calculated.

The calculations and measurements of temperature distribution in a vertical space under natural convection conditions, which are described in the literature, have generally been carried out at constant temperatures along the vertical surfaces. In a window in a building, the temperature along the vertical surfaces will vary depending on radiation and wind on the outside and on the ventilation/heating system and geometrical configuration on the inside. In consequence a vertical temperature gradient is formed along the surfaces. The calculations have therefore been supplemented with cases in which there is a vertical temperature distribution along the vertical surfaces. The assumed temperatures along the surfaces have been chosen on the basis of previous measurements on test windows in the hot box.

The experiments

The variation of heat flow along the pane has also been investigated in tests. When the temperature difference across the pane is known, the heat flow through the glass can be calculated. These values have been compared with the mathematical model.

A test arrangement has been designed and constructed at the laboratory of the Department of Building Science. In this the thermal characteristics of different window or wall constructions can be measured and studied. The test window is placed between a warm and a cold room. In the cold room, with the assistance of fans, a uniform air current is created perpendicular to the window, and

the temperature can be regulated between 0 and -30°C .

In the warm room there is a guarded hot box which is positioned around the window. The temperature difference between the box and the warm room is maintained as near zero as possible. The hot box which has an opening of 1.7×1.7 m and a depth of 0.6 m consists of 100 mm expanded polyurethane insulation sandwiched between sheets of plywood. Inside the box there is an electric tubular heater whose power consumption is measured. A vertical screen is set up in the middle of the box to prevent direct radiation between the tubular heater and the window and to maintain the natural convection inside the box. The test window is set into the heavily insulated wall (200 mm expanded polyurethane). Endeavours are made during measurements to keep all temperatures as constant as possible. In the measuring box there is an electric thermostat with on/off control of the radiator. The power supplied to keep the hot box at a constant temperature is metered. For measurement of the heat flow through the test window a correction is made with respect to the flow through the hot box and the test wall. The thermal transmittance (k) for the test window can then be calculated.

The test window consists of a 4 mm glass pane and a 6 mm sheet of plexiglass set into a frame. The air space between the panes has been varied at 10, 20, 30, 40 and 50 mm, the spacer being of wood. The width of the test window has been kept at 1200 mm and the width of the air space at 1124 mm. The height of the test window has been set at 1200, 916, 638 and 349 mm, giving air space heights of 1124, 841, 560 and 273 mm.

Along the vertical line in the centre of the window the temperature is measured on the cold surface, in the middle of the air space and on both sides of the plexiglass. In this way temperature profiles are obtained for the space and its boundaries. With knowledge of the temperature on both sides of the plexiglass and the material data (thermal conductivity, thickness), the heat flow rate at a point can be calculated.

The hot box has been calibrated using sheets of insulation material set into the opening in the test wall. The horizontal

temperature gradient at the centre of the air space has been studied in order to characterize flow. The vertical temperature profile at the centre of the air space has been studied in conjunction with transition from one flow regime to another.

When the temperature difference across the plexiglass is known, the flow can be computed. This has been illustrated for different vertical positions and different space widths and heights. This flow comprises contributions from convection+conduction and radiation, i.e. the total flow. It is seen in these calculations that the flow, a kind of local thermal transmittance, increases in proportion to the distance from the top corner. Flow is greatest at the bottom inner corner, it is here that convection on the warm side commences.

In order to study a number of air spaces, this method of measurement has been used for a test window with two 6 mm sheets of plexiglass. The size of the air space was 840x1122 mm. Along both sides of the plexiglass temperature sensors were attached so that temperatures and heat flow rates could be determined. For windows with several air spaces, the necessary number of panes was added on the cold side. The width of air space has generally been 10, 20, 40 or 80 mm - occasionally 150 mm.

In addition, different types of sealed unit supplied by Pilkington AB have been tested. Three different gases, air, Ar and SF₆, have been investigated in a single space of 6, 12 and 15 mm width. A low emissivity coating in combination with argon has also been studied in a single space. Triple glazed units (two spaces) have been investigated for an ordinary pane, with argon, and with argon in combination with a low emissivity coating.

Calculation of flows at frame/casement

In order that heat transfer at the junction between the air space in a window and the surrounding construction (wall, frame and casement if appropriate) may be studied, in-depth knowledge of geometrical configuration, constituent materials and boundary conditions is required. Heat flow through the construction will

be either two or three dimensional.

A computer program for steady two dimensional temperature distribution has been used for calculation of the temperature distribution. This program uses finite differences and demands a run time of 0.5-5 minutes on the Univac 1100 system at the Lund Computer Centre. The construction is set into an orthogonal system of coordinates, and the two axes are divided into a number of parts (max.65). In this way a cellular subdivision is obtained which can be varied along both axes. Each cell can be assigned a value of the thermal conductivity (λ), and a resistance between adjacent cells can also be stipulated. Fixed temperatures or fixed flows can be specified as boundary conditions.

Geometrical configuration has been varied in order to simulate different window constructions. The constituent materials have been varied in order that wooden, plastics or aluminium windows may be studied. The boundary conditions have been adapted to reflect the influence due to adjoining walls, different air spaces and calculated surface coefficients of heat transfer. The effects due to different spacer profiles, e.g. homogeneous materials of varying λ , and different hollow profiles of aluminium/steel/lead, have been studied. So far a total of 130 different combinations have been studied.

By comparing temperatures and heat flow rates in different cases, an idea of the way in which surface temperatures and k-values are affected can be obtained.

Further research results

Only some of the research results have been illustrated in this publication. The full results are to be found in Jonsson (1985), which is published only in Swedish.

Symbols

A	area	m^2
a	thermal diffusivity = $\lambda/\rho c_p$	m^2/s
b	width	m
c	specific heat capacity	J/kgK
c_p	specific heat capacity at constant pressure	J/kgK
c_v	specific heat capacity at constant volume	J/kgK
d	width	m
E	energy content	J/m^3
E	irradiance	W/m^2
F	angular coefficient	-
g	acceleration due to gravity	$9.81 m/s^2$
k	thermal transmittance, k-value	W/m^2K
L	height	m
L	radiance	$W/(sr \cdot m^2)$
l	distance	m
M	radiant exitance	W/m^2
m	thermal resistance	m^2K/W
m	mass	kg
n	rate	1/s
P	heat power	W
P	radiant flux, heat flow rate	W
p	surface stress	Pa
Q	quantity of heat	J
q	density of heat flow rate	W/m^2
T	fabric loss	W/K
T	thermodynamic temperature	K
T_C	cold surface temperature	K
T_H	warm surface temperature	K
t	time	s
U	voltage	V
u	velocity	m/s
V	volume	m^3
v	velocity	m/s
w	velocity	m/s

α	surface coefficient of heat transfer	$\text{W/m}^2\text{K}$
α	absorptance	-
β_h	temperature gradient in horizontal direction	-
β_v	temperature gradient in vertical direction	-
γ	cubic expansion coefficient	K^{-1}
ε	emissivity	-
φ	angle	rad
ψ	stream function	-
θ	nondimensional temperature	-
θ	angle	rad
t	temperature	$^{\circ}\text{C}$
λ	thermal conductivity	W/mK
λ	wavelength	m
ν	kinematic viscosity $=\mu/\rho$	m^2/s
μ	dynamic viscosity	Ns/m^2
ξ	vorticity	-
ρ	density	kg/m^3
ρ	reflectance	-
σ	Stefan-Boltzmann constant	$5.67 \cdot 10^{-8}$ $\text{W/m}^2\text{K}^4$
τ	shear stress	Pa
τ	transmittance	-
ω	solid angle	sr

Gr Grasshof number

Pr Prandtl number

Ra Rayleigh number

Re Reynolds number

x, y, z axes of coordinates

u, v, w velocity components in x, y, z directions

1 INTRODUCTION

1.1 BACKGROUND

Windows are the building element which did not undergo any appreciable changes for a long time. From the point of view of thermodynamics, the introduction of an inner casement during the winter was a great improvement. Since then the actual fundamental principle underlying the thermal insulation of a window, increased thermal resistance due to the introduction of an air gap between the panes of glass, has remained practically the same. Research concerning thermal insulation was from time to time neglected. It is only in recent years, because of the increased costs of heating, that an interest has been shown in improving the thermal insulation of windows.

The introduction of hermetically sealed units provided greater opportunities for the improvement of window insulation. Several air gaps (triple panes, quadruple panes, etc) have been made use of. Convective heat flow between the panes of glass can be reduced by using some other gas instead of dry air. Radiation between the panes can be reduced by giving one or more of the panes a selective coating. However, since such a coating, as a rule, also reduces the direct and diffuse radiation from the sun and the sky, energy gains will be decreased. When the energy balance across a window or in a building is studied, account must be taken both of the solar energy which is utilised and the losses of heat through the window.

Introduction of a new technology often gives rise to new problems. There may be appreciable distortion of the heat flow through the hermetically sealed unit, and calculations of heat losses and the directional operative temperature are affected. Experience of windows in Sweden has been associated with wooden windows. Wood has comparatively good thermal insulation, and the heat flow through the sealed unit is not appreciably affected by heat flow through the casement and frame. The use of other materials such as aluminium may exert considerable influence on the distribution of temperature and heat flow along the edges of the window. In addition, sealed units incorporate an aluminium spacer

which introduces a cold bridge along the edges. Even though the wooden windows have comparatively good thermal insulation, the design of the actual window construction or the incorporation of metal profiles of different kinds such as aluminium glazing fillets may change and adversely affect thermal insulation.

1.2 WINDOWS IN A BUILDING

Space heating accounts for a large proportion, about 40% of energy consumption in Sweden. According to the Statistical Year Book 1984 for Buildings and Dwellings, SCB, energy consumption in the "other" sector (heating of residential and non-residential premises etc) amounted to 157 TWh in 1983. According to Munther (1974), for an average private dwellinghouse this implies an energy consumption of 23700 kWh per annum. Losses through windows are stated to amount to 5700 kWh per annum, which is 24% of the total energy supplied. However, the building gains energy due to insolation through the windows, the magnitude of this being dependent on e.g. the size, orientation, position and shading of the glass surfaces.

In order that low annual energy consumption may be achieved in a building, the windows should be designed in such a way that the difference between heat losses and energy gains due to insolation is minimized. In this context, solar control and daylight conditions must be taken into account. It is important to make a distinction between the power requirement and the proportion of energy consumption in the building due to the window. For a southerly window, gains due to insolation during the heating season are greater than the heat losses through the window. However, during a cold winter night considerable heat losses occur. Windows with small heat flows have the following advantages:

1. they reduce the maximum power requirement of the building
2. they improve thermal climate near the window

3. they provide increased opportunities for the use of windows in easterly/westerly and northerly positions
4. they reduce the risk of condensation

1.3 LIMITATIONS, DEFINITIONS

This project is mainly concerned with heat transfer through the transparent part of a window. Through this a complex interaction between conduction, convection and radiation takes place. In addition, heat transfer during the day mainly occurs in two wavelength regions, long wave infrared radiation ($3.5\text{--}20\ \mu\text{m}$) and short wave solar radiation ($0.3\text{--}2.0\ \mu\text{m}$). Calculations and measurements in this work have been made for the hours of darkness, i.e. without short wave solar radiation. During the day a part of short wave radiation is absorbed in the glass and reduces heat losses. This may however be regarded as a secondary heat increment to the room over and above the primary transmitted daylight radiation, see Brown and Isfält (1975). The heat losses during the hours of darkness may then be taken to be valid for the entire 24 hour period.

Studies have been made of the effect due to different frame/case-ment constructions and their effect on the transparent part. Factors such as adjoining walls which may have some influence on the actual window have also been studied.

Generally speaking, across a window there exists a pressure difference which may give rise to a flow of air through the window. This flow which causes an additional heating requirement in the building has at times been included in the thermal transmittance k . This is however unfortunate since individual differences in the same window construction may give rise to completely different k -values. In addition, inlet and exhaust air flows, whether they pass through the building fabric or through ducts provided for this purpose, should form part of the ventilation system and treated as such. Consequently, inward or outward movement of air

should on no account be included in a k-value. The object of this project has been to study heat transfer through the glass assembly, casement and frame. Flow of air or infiltration through a window is not touched upon.

1.4 FURTHER RESEARCH RESULTS

Considerably more calculations and tests than referred to in the following have been carried out within this research project. They are reported in Jonsson, 1985, which is published only in Swedish.

2 HEAT TRANSFER IN A SPACE

Heat transfer through an air space consists of an interaction between the three fundamental heat transfer mechanisms, conduction, convection and radiation. In a glass assembly or in a window different materials are in addition used along the edges and give rise to a complex heat flow.

2.1 RADIATION

2.1.1 Fundamental laws

All bodies at a temperature above absolute zero, whether solids, liquids or gases, emit energy in the form of radiation and also have the capacity to absorb such radiation. This radiation is usually referred to as infrared radiation, thermal radiation or temperature radiation. When radiation takes place in a vacuum, there is no dissipation of the radiant energy. If, however, radiation passes through a gas, part of it will be absorbed. When passage through the gas is long, absorption may be considerable. This occurs when solar radiation passes through the atmosphere; there is considerable reduction in this radiation in certain wavelengths. However, absorption and emission within the infrared region (infrared radiation) by air and other gases are small for the applications to which this publication refers. Gases are therefore regarded to absorb, or to emit, no infrared radiation.

Radiance which is emitted by a black body (L_b) is distributed over different wavelengths. The spectral distribution is expressed by Planck's law (FIG.2.1.1).

$$L_{\lambda b} = \frac{2C_1}{\lambda^5 (e^{C_2/\lambda T} - 1)} \quad (2.1)$$

where $L_{\lambda b}$ = spectral distribution intensity of radiance for
a black body in a given direction (W/srm² per m)

λ = wavelength (m)

T = temperature (K)

$$C_1 = \text{constant } (0.595 \cdot 10^{-16} \text{ Wm}^2)$$

$$C_2 = \text{constant } (1.439 \cdot 10^{-2} \text{ mK})$$

Integration over all wavelengths gives

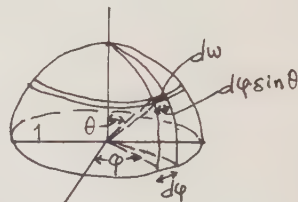
$$L_b = \int_0^{\infty} L_{\lambda b} d\lambda = \frac{\sigma}{\pi} T^4$$

The radiation in all directions within a hemisphere is termed radiant exitance M_b .

$$M_b = L_b \int_{\Delta} \cos \theta d\omega = \pi L_b$$

where ω = solid angle

θ, φ = angles according to the figure



We thus have

$$M_b = \sigma T^4 \quad (2.2)$$

where σ = Stefan-Boltzmann constant ($5.67 \cdot 10^{-8} \text{ W/m}^2 \text{K}^4$)

T = temperature (K)

The radiation which impinges on a surface can either be reflected or absorbed in this or transmitted through this, i.e.

$$\rho_{\lambda} + \alpha_{\lambda} + \tau_{\lambda} = 1 \quad (2.3)$$

where $\rho_{\lambda}, \alpha_{\lambda}, \tau_{\lambda}$ = ratio of reflected, absorbed and transmitted energy respectively to incident energy within a certain wavelength λ .

For highly absorbent media (e.g. solid bodies) transmission can be ignored, i.e. $\rho_{\lambda} + \alpha_{\lambda} = 1$. Furthermore,

$$\epsilon_{\lambda \Delta} = \alpha_{\lambda} \quad (2.4)$$

where $\epsilon_{\lambda \Delta}$ is the hemispherical spectral emissivity (see below). A body which does not reflect any radiation absorbs all incident thermal radiation and is called a black body.

The directional spectral emissivity $\epsilon_{\lambda\omega}(\lambda, \omega)$ is defined as

$$\epsilon_{\lambda\omega} = \frac{L_{\lambda}}{L_{\lambda b}} \quad (2.5)$$

where L_{λ} = spectral distribution intensity of radiance for a body in a given direction

$L_{\lambda b}$ = spectral distribution intensity for a black body

When the ratio of the radiance of a body to that of a black body according to Planck's law is the same for all wavelengths, the body is said to be grey.

Emissivity varies with the direction of radiation, and hemispherical spectral emissivity is then defined as the ratio of the monochromatic radiation of the body in question to that of a black body.

$$\epsilon_{\lambda\Delta} = \frac{M_{\lambda}}{M_{\lambda b}} = \frac{\int_{\Delta} L_{\lambda} d\lambda \cos\theta d\omega}{\int_{\Delta} L_{\lambda b} d\lambda \cos\theta d\omega} = \frac{1}{\pi} \int_{\Delta} \epsilon_{\lambda\omega} \cos\theta d\omega \quad (2.6)$$

The emissivity over all wavelengths, the total directional emissivity, is

$$\epsilon_{\omega}(\omega) = \int_0^{\infty} \epsilon_{\lambda\omega}(\lambda, \omega) \frac{M_{\lambda b}}{M_b} d\lambda \quad (2.7)$$

The special case of total normal emissivity is designated ϵ_n .

Total emissivity (ϵ_T) is defined as

$$\epsilon_T = \frac{M}{M_b} = \int_0^{\infty} d\lambda \int_{\Delta} \epsilon_{\lambda\omega} d\omega \quad (2.8)$$

The total quantity of heat which an actual body emits in the form of radiation is $M = \epsilon_T M_b$ and is thus a function of the surface temperature of the body and the structure of the surface. For a grey body, the Stefan-Boltzmann law will be

$$P = \epsilon_T \sigma A T^4 \quad (2.9)$$

where P = emitted radiation (W)

ϵ_T = total emissivity

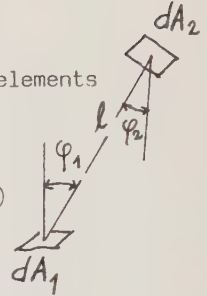
σ = Stefan-Boltzmann constant ($5.67 \cdot 10^{-8} \text{ W/m}^2 \text{K}^4$)

A = area (m^2)

T = temperature (K)

The interchange of radiation between the black surface elements dA_1 and dA_2 is

$$P_{dA_1 \rightarrow dA_2} = \frac{\sigma \cos \phi_1 \cos \phi_2}{\pi l^2} dA_1 dA_2 (T_1^4 - T_2^4) \quad (2.10)$$



where ϕ_1, ϕ_2 = angle between the ray emitted by surface 1(2) towards surface 2 (1) and the normal to surface 1(2)

l = distance between surface 1 and surface 2

The interchange of radiation between an element of surface and a surface is then

$$P_{dA_1 \rightarrow A_2} = \sigma F_{dA_1 \rightarrow A_2} dA_1 (T_1^4 - T_2^4) \quad (2.11)$$

where

$$F_{dA_1 \rightarrow A_2} = \int_{A_2} \frac{\cos \phi_1 \cos \phi_2}{\pi l^2} dA_2 \quad (2.12)$$

$F_{dA_1 \rightarrow A_2}$ is the angular coefficient for surface A_2 as seen from the element of surface dA_1 . For certain geometrical shapes and positions the angular coefficient can be solved analytically, see e.g. Eckert, Drake (1959).

Let us study a room with diffusely reflecting walls. Since the walls are not black, there will be an infinite number of reflections in the room. For surfaces which are diffuse, i.e. for which the intensity of reflected radiation is independent of direction, some expressions can be derived. The symbol q^{FROM} represents the total flux emitted by a surface per unit area and unit time.

The heat flux P_i from the surface i is

$$P_i = \sum_k A_i F_{i-k} (q_i^{\text{FROM}} - q_k^{\text{FROM}}) \quad (2.13)$$

$$P_i = A_i \frac{\epsilon_i}{1 - \epsilon_i} (\sigma T_i^4 - q_i^{\text{FROM}}) \quad (2.14)$$

where F_{ik} is the angular coefficient for surface k seen from surface i .

These equations, it is generally P_i and q_i^{FROM} which are the unknowns. For two parallel infinite surfaces $i=2$ four unknowns and four equations are obtained

$$P_1 = A(q_1^{\text{FROM}} - q_2^{\text{FROM}})$$

$$P_2 = A(q_2^{\text{FROM}} - q_1^{\text{FROM}})$$

$$P_1 = A \frac{\epsilon_1}{1 - \epsilon_1} (\sigma T_1^4 - q_1^{\text{FROM}})$$

$$P_2 = A \frac{\epsilon_2}{1 - \epsilon_2} (\sigma T_2^4 - q_2^{\text{FROM}})$$

By eliminating q_1^{FROM} , q_2^{FROM} and P_2 , we have

$$P_1 = \frac{\sigma A (T_1^4 - T_2^4)}{1/\epsilon_1 + 1/\epsilon_2 - 1} \quad (2.15)$$

If a surface A_1 is surrounded by another surface A_2 such that each part of A_2 can be seen from A_1 , we have

$$P_1 = \frac{\sigma A_1 (T_1^4 - T_2^4)}{1/\epsilon_1 + A_1/A_2 (1/\epsilon_2 - 1)} \quad (2.16)$$

2.1.2 Radiation characteristics

Most solid substances absorb practically all incident radiation in a very thin layer below the surface. For metals and electrically non-conductive materials, the thickness of this layer is a fraction of μm . For silver, gold and copper a layer thickness of approximately $0.025 \mu\text{m}$ is required for the transmitted solar radiation to be reduced to one tenth ($\lambda=0.7-2 \mu\text{m}$). Some solid materials such as glass, quartz and most liquids constitute an exception for the visible region and some of the infrared region. Further down in the infrared region absorption is very intensive. Arsensulfid glass has been developed for its good transmission characteristics within the infrared region (FIG.2.1.2).

It is seen from FIG.2.1.3 that emissivity varies considerably in different directions. Emissivity also varies with surface conditions and temperature. The ratio of hemispherical radiation to that in the direction of the normal varies between 0.95 for non-conductive materials and 1.25 for bright metals, according to Eckert (1972).

Lohrengel (1970) has measured the emissivity of different kinds of glass within the $1.0-23 \mu\text{m}$ wavelength region. This increased with temperature but was practically constant from -30°C to $+30^\circ\text{C}$. The ratio of the emissivity for hemispherical radiation (ϵ_A) to that for radiation in the direction of the normal (ϵ_n) was 0.935 for window glass and 0.915 for silica glass. The values obtained for window glass were $\epsilon_A=0.853$ and $\epsilon_n=0.92$ and for silica glass $\epsilon_A=0.78$ and $\epsilon_n=0.855$.

For windows there are two wavelength regions in particular which are of interest, namely infrared radiation at 6000 K, corresponding to the solar spectrum at the surface of the earth ($0.3-2.0 \mu\text{m}$) and the long wave infrared radiation at 293 K where the wavelengths cover the region from 3.5 to $>20 \mu\text{m}$.

For the first of the above regions the transmission characteristics of the glass can be varied in order to regulate insolation (solar control). For the long wave region the heat losses can be changed. For a thermally economic window the insolation should be

a maximum and the heat loss a minimum, i.e. for $\lambda=0.3$ - approx. $2.0\ \mu\text{m}$ transmission (τ) should be practically 100% and reflection (ρ) practically 0%, and for λ = approx 3.5 - $20\ \mu\text{m}$, τ should be practically 0% and ρ practically 100% (see FIG.2.1.4). Layers which have different properties in different wavelength regions are called selective layers. A window can be optimized from the point of view of thermal economy by giving it selective transmission characteristics.

For a window to have the above characteristics, the change-over point between transmission and reflection should be at about $2.5\ \mu\text{m}$. On the other hand, a window with thermal solar control should retain its transmission for visible light and reflect the greatest proportion of the infrared radiation from the sun, i.e the change-over point should be at about $0.8\ \mu\text{m}$.

Ribbing (1982) lists three physical mechanisms whereby a selective layer can be provided,

- a) interference in alternating multiple metallic and dielectric layers
- b) change between reflectance and transmittance in mobile charged particles
- c) geometrical design of the coating

The first effect makes possible a very precisely defined change-over in reflectance characteristics, but diffusion between the layers in a multiple layer coating limits the choice of materials. There are great practical difficulties in obtaining a uniform and well defined layer thickness.

The effect according to b) is due to freely moving charged particles which reflect long wave electromagnetic radiation but change at a certain wavelength to transmit radiation with very little attenuation. For a metal the conduction electrons are very mobile over large volumes, which means that the density is too large and the limiting wavelength for the change-over is not greater than about $0.5\ \mu\text{m}$.

In semiconductors the density of conduction electrons is smaller than in metals and can be controlled by doping, the addition of

impurities, so that the limiting wavelength can be made to approach the ideal curve.

The effect according to c) is obtained by coating glass with a metallic grating of such dimensions that the apertures are large in relation to the wavelength of visible light and small in relation to the wavelength of infrared radiation.

In reality the different mechanisms may interact. For the layer structure according to 1) and 2) below, interaction which occurs between mechanisms a) and b) is utilized. The methods mostly employed so far for the construction of a selective filter have been

- 1) a thin layer of a noble metal (Ag, Au, Cu)
- 2) a multiple layer constructed about a thin metal film given anti-reflection treatment, consisting of dielectric layers (e.g. ZnS/Ag/ZnS) and layers of different types of semiconductor (e.g. In_2O_3 , SnO_2)

According to Ribbing (1982), measurements on multilayer films have shown that noble metals in the order Cu, Ag, Au have the most favourable properties. Since stannic oxide has many suitable properties for a selective filter, a lot of development work has been done on this semiconductor. Karlsson, Roos, Ribbing (1983) carried out tests on SnO_2 doped with NH_4F . This layer has shown very promising characteristics, see FIG.2.1.5.

2.2 CONDUCTION

2.2.1 Fundamental laws

Flow of heat in a body takes place from a higher towards a lower temperature. Points in a body which have the same temperature at a certain point in time form a surface called an isothermal surface. The distribution of temperature in a body can thus be illustrated by means of a number of isothermal shells of different thickness.

At a point in a body, heat flux can generally be defined as

$$\frac{P}{A} = -\lambda \frac{\partial \vartheta}{\partial n} \quad (2.17)$$

This relationship is known as the Fourier law. The heat flux (P) moves in the direction of the normal to the surface (n) towards a lower temperature, producing a negative temperature gradient. This relationship is generally transformed for one-dimensional steady heat transfer to

$$P = \frac{\lambda A (\vartheta_1 - \vartheta_2)}{d} \quad (2.18)$$

where d is the thickness of the layer and A its area, and ϑ_1 and ϑ_2 the surface temperatures.

Thermal conductivity (λ) is generally assumed to be a material constant but in actual fact it is a function of temperature and, for gases and liquids, a function of pressure. However, for glass the temperature dependence of thermal conductivity is so slight that it can be ignored. For some materials (e.g. wood), thermal conductivity is directional.

Generally speaking, the material properties thermal conductivity λ , specific heat capacity c_p and density ρ may be described as functions of x , y , z and ϑ .

The material is generally assumed to be isotropic, i.e. properties at a point inside the material are the same irrespective of the direction selected.

If we consider an infinitely small control volume of the dimensions dx , dy and dz in a three dimensional system of coordinates, the energy balance for this control volume gives rise to the equation

$$\rho c_p \frac{\partial T}{\partial t} = \frac{\partial}{\partial x} \left(\lambda \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(\lambda \frac{\partial T}{\partial y} \right) + \frac{\partial}{\partial z} \left(\lambda \frac{\partial T}{\partial z} \right) + P \quad (2.19)$$

where P = heat generated inside the control volume per unit volume and time

The equation is valid for an isotropic heterogeneous body where λ , c_p and ρ are independent of temperature.

If there are no heat sources and the material is homogeneous, i.e. λ is constant, the equation is simplified to

$$\frac{\partial T}{\partial t} = \frac{\lambda}{\rho c_p} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) = a \nabla^2 T \quad (2.20)$$

where $a = \lambda / \rho c_p$ is thermal diffusivity.

For onedimensional steady heat flow, $\partial T / \partial t = 0$. The equation is then

$$a \frac{\partial^2 T}{\partial x^2} = 0 \quad (2.21)$$

which implies linear temperature distribution.

For solids and liquids, $c_p \approx c_v$.

The differential equation for thermal conduction can be solved either analytically or by the use of numerical methods.

Flow of heat at a surface involves an interaction between convection and conduction. This heat transfer mechanism will be discussed fully in the section on convection. A simple expression can be derived by making use of certain similarities between convection and conduction. If we study a fluid flowing over a surface, temperature distribution will be approximately as shown in FIG. 2.2.1. For a thin layer immediately next to the surface, heat transfer will take place by pure conduction. Further away from the surface convection will exert increasing influence on heat transfer. We can define a layer δ' where heat transfer is supposed to occur only by conduction. If the thermal conductivity of the fluid is λ_F , the Fourier equation for heat transfer at the surface will have the form

$$P = \frac{\lambda_F A (\vartheta_S - \vartheta_F)}{\delta} \quad (2.22)$$

If δ' is known, P can be calculated. However, δ' is a function of e.g. flow conditions, the shape and structure of the surface. In practice it has been found appropriate to determine λ_F / δ' which is designated the convective surface coefficient of heat transfer (α_c).

2.3 CONVECTION

2.3.1 Fundamental concepts

2.3.1.1 Laminar and turbulent flow

Transfer of heat between a solid surface and an adjacent flowing fluid takes place through a combination of conduction and energy transfer due to movement of the fluid. Each particle is assigned a certain energy content which is determined by its temperature and specific heat capacity. If the temperature of the surface is higher than that of the fluid, heat will flow by conduction from the solid surface to those fluid particles which are in the vicinity of the surface. This heat flux increases the internal energy

of the fluid and is carried away by its motion. When the particles of fluid reach a region at a lower temperature, heat is transferred to the solid surface by conduction. This energy transfer is called convection. Transfer of heat by conduction and convection between a surface and a fluid may be expressed by the convective surface coefficient of heat transfer (α_c). See equation (2.22).

$$\alpha_c = \frac{P_c}{A(\vartheta_s - \vartheta_f)} \quad (2.23)$$

where

- P_c = heat transfer by conduction and convection (W)
- A = area (m^2)
- α_c = convective surface coefficient of heat transfer (W/m^2K)
- $\vartheta_s - \vartheta_f$ = temperature difference (K) between a surface and a fluid

Since convection is so closely associated with the motion of the fluid, it is necessary to know about the mechanism of fluid motion.

The forces which act on the fluid may be body forces (e.g. gravitational forces) or surface forces (e.g. shear and compressive forces). If the fluid is set in motion by body forces, which may occur due to variations in density under the influence of gravitational forces, the phenomenon is called natural convection. These variations in density occur due to differences in temperature, for instance when the fluid is heated. If the convection process is initiated by surface forces, for instance due to a pump or fan, it is referred to as forced convection.

Convective flow may be laminar or turbulent. The difference between laminar and turbulent flow is manifested by the appearance of the streamlines in the fluid, i.e. the path taken by a particle. The term particle in this context does not refer to individual molecules but a part of the fluid of such size that the mass

inside this may be assumed to be continuous.

Laminar flow is characterized by parallel streamlines of uniform and continuous appearance. On the other hand, turbulent flow is characterized by the fact that the particles move about in a disordered manner, they form wakes, move in a zig-zag pattern and in an irregular fashion. However, flow as a whole may be uniform and regular.

During laminar flow along a surface which has a temperature different from that of the fluid, heat will be transferred by molecular conduction within the fluid and between the fluid and the surface. There is no mixing in the flow which would cause the energy of the particles to be transmitted across the streamlines.

During turbulent flow, owing to the innumerable wakes and eddies, the particles will be transported right across the mean streamlines. By virtue of the fact that they mix with other particles, these particles then act as energy carriers. If mixing (turbulence) in the fluid increases, transfer of heat by convection will therefore increase. The velocity component u in turbulent flow can be written as $u = \bar{u} + u'$ where $\bar{u}$ is the mean with respect to time and u' represents the instantaneous variations in u . A steady flow in which $\bar{u}$ does not change during the period can be defined in this way. It is generally the flow field characterized by the mean velocity with respect to time which is the principal feature of turbulent flow and it is therefore this which is usually of interest.

As stated above, the motion of a fluid is determined by the forces acting on the particles, i.e. body forces (gravitation, electrical or magnetic forces, centrifugal forces) or surface forces (shear forces, compressive forces). Stresses due to surface forces acting on a fluid are related to the time function of deformation. An expression given for this deformation by Newton is

$$\tau = \mu \frac{du}{dy}$$

where

μ = dynamic viscosity (Ns/m^2)

$\frac{du}{dy}$ = velocity gradient perpendicular to the
velocity vector (m/(sm))

This means that between two planes in the direction of flow there is a shear stress τ . A fluid which conforms to the above relationship is called a newtonian fluid. Generally speaking, the viscosity of a gas is relatively small. This means that shear forces are large only when the velocity gradient is steep. If the shear forces are ignored when the velocity gradient is slight, a simplified description of convection is obtained. The actual fluid is replaced by an idealized model, an inviscid gas. Since viscous flow is associated with shear forces and thus internal friction, it follows that an inviscid fluid has no internal friction. This is obviously quite unrealistic, but in certain cases the assumption is admissible since it very considerably simplifies mathematical description. It is sometimes possible to use potential theory.

A boundary layer is characterized by the fact that a certain variable, for instance the temperature or velocity, exhibits large variations within a limited region. If a plate of different temperature is placed in a flowing medium, a velocity boundary layer and a temperature boundary will be formed along the surface of the plate.

At the surface of the plate velocity is zero. At some distance from the surface it is equal to the undisturbed free velocity. Between these two points the velocity changes from zero to the full value. The thickness of the boundary layer can be defined as the region in which the velocity variations occur. Outside the boundary layer only small variations in temperature and velocity occur, and the shear forces are therefore small and can be ignored. Flow here is regarded as inviscid. This subdivision into boundary layers and inviscid regions simplifies the general equations of flow.

The appearance of the velocity profile and the boundary layer depends on the character of the flow. We will study a plate where

flow is parallel to the surface of the plate (FIG.2.3.1). Since some of the particles in the fluid hit the edge of the plate ($x=0$), these will be retarded while the others will remain unaffected. As the fluid moves along the plate, shear forces will act on the flow, and consequently an increasing portion of the flow will be affected. The boundary layer therefore increases along the plate from zero at the leading edge. Near the edge of the plate the velocity profile represents laminar flow. This flow pattern persists up to a certain distance from the edge of the plate, beyond this the character of flow changes and becomes turbulent. Between these two different flow patterns there is a transition zone. Where the transition between laminar and turbulent flow occurs can be approximately predicted with the assistance of Reynolds Number (Re) - see below. Since l which is the characteristic length in Re represents the distance from the edge of the plate, Re increases in proportion to l . The value of Re at which turbulence occurs is dependent on surface contour, surface roughness, the level of disturbance and even heat transfer. The critical Reynolds Number can vary between $8 \cdot 10^4$ and $2 \cdot 10^6$, but is generally around $5 \cdot 10^5$. There are always small wakes in the flow, but so long as the viscous forces are large their growth is prevented. As the laminar boundary layer increases in thickness, the viscous forces gradually decreases in significance in relation to inertia forces. The disturbances will then increase, causing the boundary layer to become unstable and a transition to turbulent flow to occur. The laminar boundary layer is destroyed, but a quasi-laminar boundary layer remains in the form of a thin film along the solid surface and is designated laminar sublayer. Between this layer and the turbulent core there is a buffer layer (FIG.2.3.2).

2.3.1.2 Nondimensional numbers

By using nondimensional numbers, characteristics according to a certain flow model can be summarized. These characteristics may refer to the type of flow, flow conditions or heat transfer mechanism. In this way it is possible to compare or predict the results obtained from a certain (theoretical or actual) experimental arrangement with the results obtained from another

arrangement.

The equations which describe flow can, by various transformations, be converted into a system of equations containing only nondimensional variables and numbers. Apart from the variables temperature, time, velocity and length (x, y, z coordinates), the equations may contain nondimensional quantities such as the Grashof number (Gr) and Prandtl number (Pr). When two experimental arrangements are compared, it is essential that the nondimensional numbers should be defined in the same way. Different boundary conditions will also affect the solutions.

Reynolds number

$$Re = \frac{vl}{\nu} \quad (2.24)$$

where v and l are characteristic velocity and length, and $\nu = \mu/\rho$ is the kinematic viscosity (m^2/s).

The Reynolds number can be transformed into

$$Re = \frac{\rho v^2 l}{\mu}$$

This means that Re denotes the relationship between inertia forces and viscous forces - see the terms relating to inertia and viscosity in equation (2.43). For low values of Re the viscous forces are large in relation to inertia forces. Since the viscous forces exert a stabilizing influence on flow, flow will be laminar. As Re increases the viscous forces gradually lose their significance as stabilizing forces. Via a transition phase, flow will change from laminar to turbulent.

Prandtl number (Pr)

$$Pr = \frac{\nu}{a} = \frac{c_p \mu}{\lambda} \quad (2.25)$$

where

$$\nu = \frac{\mu}{\rho} = \text{kinematic viscosity}$$

$$\alpha = \frac{\lambda}{\rho c_p} = \text{thermal diffusivity}$$

The Prandtl number describes the material characteristics of the flow medium. Since ν is a measure of the momentum which influences velocity distribution and α is thermal diffusivity which influences the temperature profile, Pr is the relationship between a mechanical characteristic and a thermal conduction characteristic. This is made use of in studies of boundary layers since $\sqrt{Pr}$ denotes the relationship between the hydrodynamic boundary layer (according to the velocity profile) and the thermodynamic boundary layer (according to the temperature profile), Gebhart (1961).

Grashof number (Gr)

$$Gr = \frac{g \beta \Delta T d^3}{\nu^2} \quad (2.26)$$

where ΔT is the characteristic temperature difference and d the characteristic distance, and β the cubic expansion coefficient (K^{-1}).

Gr is part of the force equation in the nondimensional system of equations which describes natural convection, and represents the relationship between buoyant forces and viscous forces. As Gr increases, the buoyant forces which start and maintain the convection process increase in relation to the viscous forces. This means that heat transfer by convection increases in relation to heat transfer by conduction. When buoyant forces are the only motive force, the velocity of flow is determined only by Gr . In natural convection Re is replaced by Gr .

Rayleigh number (Ra)

$$Ra = Pr \cdot Gr = \frac{g \gamma \Delta T d^3}{\nu_a} = C_{fluid} \Delta T d^3 \quad (2.27)$$

where

γ = cubic expansion coefficient, where $\gamma = 1/T_m$
is valid for gases

T_m = mean temperature (K)

$\Delta T, d$ = characteristic temperature difference and
distance respectively

The constant C_{fluid} is the property of the fluid and varies with the mean temperature. TAB.2.3.1 contains thermodynamic data for air at different temperatures.

At 0 °C, $C_{fluid} = 0.144 \cdot 10^9$ for air. For other gases see also TAB.3.6.1.

If ΔT and T_m are put at 20 K and 273 K respectively and the fluid is air, the values of the Rayleigh number for different widths d of a vertical air space are as follows

$d(\text{mm})$	Ra
6	600
12	5000
20	23000
30	78000
40	184000

In conjunction with heat transfer from a solid surface to a flowing medium by means of convection, the surface heat transfer is a function of Ra and Pr according to the system of equations for natural convection (e.g. equation (2.47)).

If Ra increases, viscous forces will exert an increasingly small influence on the balance of forces in the medium. In this way it

is possible with the aid of Ra to characterize three regimes for convection in a vertical space, namely the conduction, transition and boundary layer regimes, Eckert, Carlsson (1961). In addition, McGregor (1969) introduced a transition regime and a turbulent boundary layer regime in which secondary or tertiary flows occur.

Nusselt number (Nu)

The Nusselt number is defined as

$$Nu = \frac{\alpha d}{\lambda} = \frac{\alpha}{\lambda/d} \quad (2.28)$$

where

α = convective surface coefficient of heat transfer

d = characteristic length

λ = thermal conductivity of stationary fluid

This means that Nu expresses the ratio of heat flow by convection and conduction to the heat flow by conduction only, across a layer of thickness d . Heat transfer at the surface is characterized by a nondimensional number Nu.

2.3.2 Fundamental equations

2.3.2.1 General

Transfer of heat in a fluid usually consists of an interaction between convection and conduction. Where it is body forces that create motion in the system where transfer takes place, this is called natural convection.

Simultaneously with convective heat transfer there is energy transfer, and for this reason the laws of mechanics and thermodynamics must be applied. The problem can be tackled either by studying the individual particles and describing their motion in statistical terms, or by describing the motion of a large number of particles and applying the fundamental laws of mechanics and thermodynamics. The latter approach is the easier one since it disregards the individual particles and instead studies the fluid as a whole. If the fluid consists of a few particles within a limited volume, e.g. gases at low pressure, this method cannot be applied as the condition stipulating a very large number of particles is not satisfied. In order that a system of equations may be set up, the convection process must first be described with the assistance of the laws of mechanics and thermodynamics. In this way a theoretical model can be constructed which can be described in mathematical terms.

In the theoretical model, a control volume of the fluid which is separated from the rest of the fluid by control sides is examined. It is considered that this control volume is fixed in relation to a system of coordinates, with the fluid flowing through the control surfaces. If information concerning local quantities is to be obtained, a very small control volume must be selected, but it must not be so small that the particle structure becomes significant. This can be illustrated in the case of density (ρ) which is defined as $\rho = \Delta m / \Delta V$, where Δm is the mass inside the control volume ΔV . If the control volume is made large, then the density represents a mean value. As the control volume is reduced in size, density will increasingly refer to a local value. If the volume is reduced to such an extent that only a few particles are contained inside it, the loss or addition of a particle will have

a very great influence on density and this will undergo large fluctuations. In the continuity model it is assumed that density attains an asymptotic value before these fluctuations occur. This value is called local density. The following derivation of the equations relating to continuity, momentum and energy are based mainly on Eckert (1972).

2.3.2.2 Continuity equation

This equation describes the law of the conservation of matter. We study a control volume of dimensions dx , dy and dz (FIG.2.3.3) in a system of coordinates. Since matter cannot be created or destroyed, the difference between the mass flow rate into and out of the control volume must be equal to the increase of mass within the volume. The mass per unit time which passes through surface 1 and into the control volume is then $\rho u dy dz$, where ρ is the density.

The difference between the mass flows entering and leaving the control volume is

$$\frac{\partial(\rho u)}{\partial x} dy dz dx \quad \text{in the direction of } x$$

$$\frac{\partial(\rho v)}{\partial y} dx dz dy \quad \text{in the direction of } y$$

$$\frac{\partial(\rho w)}{\partial z} dx dy dz \quad \text{in the direction of } z$$

The net increase of matter per unit time is

$$\frac{\partial \rho}{\partial t} dx dy dz$$

The law of the conservation of matter gives rise to the equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \quad (2.29)$$

2.3.2.3 Momentum equation

This equation states that the sum of all forces acting on the mass must be equal to the time rate of change of the impulse ($F = \frac{d}{dt} (mv)$).

We define that the body forces, e.g. gravitation, act on the fluid in the control volume with the component g_x in the direction of x . The surface forces on a surface are broken down into a component at right angles to the surface and two components parallel to the surface. These forces per unit area are called normal and shear stresses respectively, e.g. the stress p_{xy} will act parallel to the y axis on a surface which is perpendicular to the x axis. Positive stresses are defined according to FIG 2.3.3. The stresses shall decrease as the velocity of the gas decreases, since the shear forces for a fluid at rest are zero. The normal surface forces are transformed into $p_{xx}-p$, $p_{yy}-p$ and $p_{zz}-p$, where p =fluid pressure.

Let us study in the control volume the forces which act in the x , y and z directions.

The momentum equations can be written

$$\begin{aligned} \oint \frac{Du}{Dt} &= -\frac{\partial p}{\partial x} + \frac{\partial p_{xx}}{\partial x} + \frac{\partial p_{yx}}{\partial y} + \frac{\partial p_{zx}}{\partial z} + \oint g_x \\ \oint \frac{Dv}{Dt} &= -\frac{\partial p}{\partial y} + \frac{\partial p_{xy}}{\partial x} + \frac{\partial p_{yy}}{\partial y} + \frac{\partial p_{zy}}{\partial z} + \oint g_y \\ \oint \frac{Dw}{Dt} &= -\frac{\partial p}{\partial z} + \frac{\partial p_{xz}}{\partial x} + \frac{\partial p_{yz}}{\partial y} + \frac{\partial p_{zz}}{\partial z} + \oint g_z \end{aligned} \quad (2.30)$$

where

$$\frac{D}{Dt} = \frac{\partial}{\partial t} + u \frac{\partial}{\partial x} + v \frac{\partial}{\partial y} + w \frac{\partial}{\partial z}$$

This is the general momentum equation which has a very all-embracing validity since it is based on the fundamental laws of mechanics. In order that the equation may be simplified, a relationship between the surface forces and the deformation of the fluid particles must be constructed. For a newtonian fluid there is a linear relationship between shear stress (τ) and deformation ($\partial u / \partial y$) as expressed by $\tau = \mu \partial u / \partial y$. Stokes derived the following general relationship on the assumption that the surface forces are proportional to the rate of deformation.

$$P_{xy} = P_{yx}$$

$$P_{yx} = \mu \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \right)$$

$$P_{xx} = 2\mu \frac{\partial u}{\partial x} - \frac{2}{3}\mu \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right)$$

With the aid of the Kronecker delta (δ_{ij}), the relationship can be written

$$P_{ij} = \mu \left(\frac{\partial v_i}{\partial x_j} + \frac{\partial v_j}{\partial x_i} \right) - \frac{2}{3}\mu \sum_k \frac{\partial v_k}{\partial x_k} \delta_{ij} \quad (2.31)$$

där

$$\delta_{ij} = \begin{cases} 1 & \text{for } i=j \\ 0 & \text{for } i \neq j \end{cases}$$

The continuity and momentum equation can now be written as

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u)}{\partial x} + \frac{\partial(\rho v)}{\partial y} + \frac{\partial(\rho w)}{\partial z} = 0 \quad (2.32)$$

$$\begin{aligned} \rho \frac{Du}{Dt} = & -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x} \left[2\mu \frac{\partial u}{\partial x} - \frac{2}{3}\mu \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right] + \\ & + \frac{\partial}{\partial y} \left[\mu \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \right) \right] + \frac{\partial}{\partial z} \left[\mu \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right) \right] + \rho g_x \quad (2.33) \end{aligned}$$

$$\begin{aligned} \varrho \frac{Dv}{Dt} = & -\frac{\partial p}{\partial y} + \frac{\partial}{\partial y} \left[2\mu \frac{\partial v}{\partial y} - \frac{2}{3}\mu \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right] + \\ & + \frac{\partial}{\partial x} \left[\mu \left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y} \right) \right] + \frac{\partial}{\partial z} \left[\mu \left(\frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \right) \right] + \varrho g_y \quad (2.34) \end{aligned}$$

$$\begin{aligned} \varrho \frac{Dw}{Dt} = & -\frac{\partial p}{\partial z} + \frac{\partial}{\partial z} \left[2\mu \frac{\partial w}{\partial z} - \frac{2}{3}\mu \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right) \right] + \\ & + \frac{\partial}{\partial x} \left[\mu \left(\frac{\partial w}{\partial x} + \frac{\partial u}{\partial z} \right) \right] + \frac{\partial}{\partial y} \left[\mu \left(\frac{\partial w}{\partial y} + \frac{\partial v}{\partial z} \right) \right] + \varrho g_z \quad (2.35) \end{aligned}$$

These four equations are the Navier-Stokes equations. The right hand sides of the equations represent the motive forces while the left hand sides are the product of mass and acceleration ($\varrho Du/dt$)

For a medium of constant properties, where ϱ and μ are independent of time and space, the equations are simplified so that four unknown variables (u, v, w and p) and four equations remain. In this way it is possible to solve the system of equations on the basis of the boundary conditions. The condition of constant properties is satisfied for a model representing liquids in which there are moderate temperature differences, and gases where the pressure variations are small. Since the equations can be solved irrespective of the temperature in the medium, this means that the flow is the same whether the temperature is the same all over the medium or whether a temperature field is set up owing to heat transfer. If the constant properties are satisfied in a certain calculation case, the Navier-Stokes equations can be solved to calculate the flow conditions. The temperature distribution can then be solved and superimposed on the flow field. In calculating natural convection, these simplifications cannot be made since density (ϱ) is a function of temperature and thus varies in time and space. It is after all, these variations in density that give rise to natural convection.

2.3.2.4 Energy equation

The energy equation is an expression of the law of the conservation of energy, i.e. that the sum of the net energy passing through the control surfaces must be equal to the increase in energy content within the control volume. The energy balance can then be written

$$dE_i = dE_c + dE_{\text{cond}} + dE_w \quad (2.36)$$

where

dE_i = accumulation of internal energy and kinetic energy ($\text{J/m}^3\text{s}$)

dE_c = net increase in internal and kinetic energy owing to convection

dE_{cond} = net increase in heat owing to conduction

dE_w = net work done by the surroundings on the system

Since the equation holds for an arbitrary instant, each term refers to the change in energy per unit time. The kinetic energy can be related to the velocity of flow, while the internal energy depends on the internal motion of molecules, i.e. it is a function of local temperature and density in the fluid.

According to Bird, Stewart, Lightfoot (1960), for a newtonian fluid of constant conductivity the equation is

$$\rho c_v \frac{DT}{Dt} = \lambda \nabla^2 T - T \left(\frac{\partial \rho}{\partial T} \right)_V (\nabla \cdot \vec{v}) + \mu \phi_V \quad (2.37)$$

On the right hand side the terms denote change in internal energy due to conduction, expansion effects (reversible) and irreversible viscous heating. The term ϕ_V is usually referred to as viscous dissipation (heat of friction).

If we ignore the term ϕ_V , and ρ for the fluid is independent

of T , we have

$$\rho c_p \frac{DT}{Dt} = \lambda \nabla^2 T$$

$$\frac{\partial T}{\partial t} = -u \frac{\partial T}{\partial x} - v \frac{\partial T}{\partial y} - w \frac{\partial T}{\partial z} + a \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \quad (2.38)$$

where

$$a = \lambda / \rho c_p, \text{ thermal diffusivity}$$

The right hand side of the equation expresses the net quantity of heat supplied, the first three terms that due to convection and the last three that due to conduction, which gives rise to the increase in temperature according to the left hand side. Two terms, relating to heat of compression and heat of friction, are ignored.

For a solid body the velocity is $v=0$, and the expression becomes

$$\rho c_p \frac{\partial T}{\partial t} = \lambda \nabla^2 T \quad (2.39)$$

which is the equation of thermal conduction in a solid material.

In the three dimensional case five coupled differential equations must be solved. In general, there are no exact solutions, and numerical iterations must be used.

2.3.3 Natural convection

2.3.3.1 General

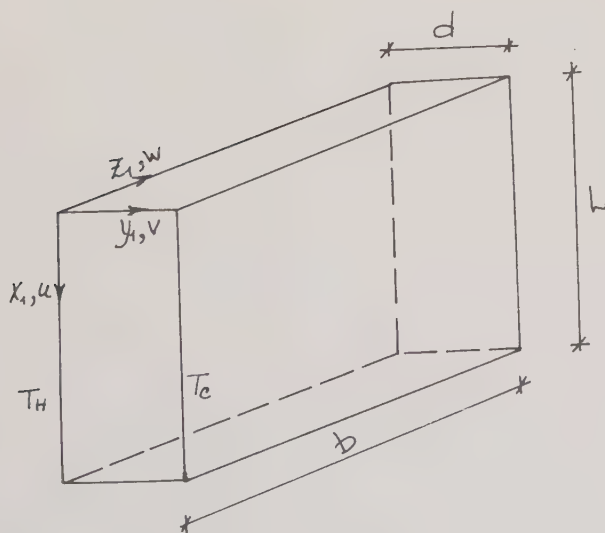
In broad outline, the problem may be said to be one of heat transfer, from the plate to the fluid and internally between the plates. In natural convection a boundary layer is created along the edge surfaces. The inner region, the core, is completely surrounded by boundary layers. For this reason the inner region and

the boundary layer cannot be completely independent of one another but must be closely coupled. Analytical solution of the problem is therefore difficult.

The two heat transfer mechanisms, radiation and conduction + convection, naturally take place simultaneously within a space. However, radiation is independent of conduction and convection. If the temperatures and radiation properties of the boundary surfaces are known, radiation can be calculated in an acceptable manner.

2.3.3.2 Natural convection between vertical plates

The general equations proposed in subsection 2.3.2 must be simplified in order that natural convection may be treated. The space is assumed to consist of two parallel vertical surfaces bounded by two horizontal planes. The system of coordinates (x_1 , y_1 , z_1) and the velocity (u, v, w) are denoted as shown below.



In the remaining direction the space is supposed to be of infinite extent, i.e. the problem is two dimensional. Let T_H and T_C be the temperatures of the vertical surfaces and the distances d and L the width and height respectively of the space. Let us assume that the fluid is incompressible ($\nabla \cdot \vec{v} = 0$) and that the heat of friction can be ignored. Since the difference in temperature between the warm and the cold side is small in comparison with the thermodynamic temperature, the properties of the fluid may be assumed to be constant apart from the variations in density which give rise to convection. For the steady case, the continuity equation will then be (see also equation (2.32)).

$$\frac{\partial u}{\partial x_1} + \frac{\partial v}{\partial y_1} = 0 \quad (2.40)$$

For the steady case, the momentum equation according to (2.32)-(2.35) is simplified to

$$\underbrace{\oint \left(\frac{\partial u}{\partial t} \right)}_{=0} + u \frac{\partial u}{\partial x_1} + v \frac{\partial u}{\partial y_1} = - \frac{\partial p}{\partial x_1} + \mu \left(\frac{\partial^2 u}{\partial x_1^2} + \frac{\partial^2 u}{\partial y_1^2} \right) + \underbrace{\mu \frac{\partial}{\partial x_1} \left(\frac{\partial u}{\partial x_1} + \frac{\partial v}{\partial y_1} \right)}_{=0} + \rho g_{x_1}$$

$$\rho \left(u \frac{\partial v}{\partial x_1} + v \frac{\partial v}{\partial y_1} \right) = - \frac{\partial p}{\partial y_1} + \mu \left(\frac{\partial^2 v}{\partial x_1^2} + \frac{\partial^2 v}{\partial y_1^2} \right) \quad (2.41)$$

The cubic expansion coefficient is defined as

$$\gamma = \frac{1}{V} \left(\frac{\partial V}{\partial T} \right)_P = - \frac{1}{\rho} \left(\frac{\partial \rho}{\partial T} \right)_P$$

and with T_C as the reference level, we have

$$\vartheta(T) = \vartheta(T_C) - \vartheta(T_C) \chi(T-T_C)$$

and

$$\vartheta_{g_{x_1}} = g \vartheta(T_C) (1 - \chi(T-T_C))$$

and the momentum equation becomes

$$u \frac{\partial u}{\partial x_1} + v \frac{\partial u}{\partial y_1} = -\frac{1}{\vartheta} \frac{\partial p}{\partial x_1} + g$$

$$-g \left(\frac{T-T_C}{T_C} \right) + \vartheta \left(\frac{\partial^2 u}{\partial x_1^2} + \frac{\partial^2 u}{\partial y_1^2} \right) \quad (2.42)$$

$$u \frac{\partial v}{\partial x_1} + v \frac{\partial v}{\partial y_1} = -\frac{1}{\vartheta} \frac{\partial p}{\partial y_1}$$

$$+ \vartheta \left(\frac{\partial^2 v}{\partial x_1^2} + \frac{\partial^2 v}{\partial y_1^2} \right) \quad (2.43)$$

Equation (2.42) is differentiated by $\partial/\partial y_1$ och equation (2.43) by $\partial/\partial x_1$.

The term $\frac{1}{\vartheta} \frac{\partial^2 p}{\partial x_1 \partial y_1}$

can be eliminated by subtracting the equations.

The energy equation according to (2.38) becomes

$$u \frac{\partial T}{\partial x_1} + v \frac{\partial T}{\partial y_1} = a \left(\frac{\partial^2 T}{\partial x_1^2} + \frac{\partial^2 T}{\partial y_1^2} \right) \quad (2.44)$$

where $a = \lambda / \vartheta c_p$

In order that the variables may be made nondimensional, let $x=x_1/d$ and $y=y_1/d$.

The continuity equation is satisfied by the stream function ψ as

$$u = \frac{a}{d} \frac{\partial \psi}{\partial y} \quad v = -\frac{a}{d} \frac{\partial \psi}{\partial x} \quad (2.45)$$

$$\theta = \frac{T - T_C}{T_H - T_C} \quad (2.46)$$

which yields, according to Ostrach (1972)

$$\nabla^4 \psi = Ra \frac{\partial \theta}{\partial y} + \frac{1}{Pr} \left[\frac{\partial}{\partial x} (\nabla^2 \psi) \frac{\partial \psi}{\partial y} - \frac{\partial}{\partial y} (\nabla^2 \psi) \frac{\partial \psi}{\partial x} \right] \quad (2.47)$$

$$\nabla^2 \theta = \frac{\partial \psi}{\partial y} \frac{\partial \theta}{\partial x} - \frac{\partial \psi}{\partial x} \frac{\partial \theta}{\partial y} \quad (2.48)$$

At the vertical and horizontal edges the velocity of the fluid (u and v) is zero, i.e. $\partial \psi / \partial y = 0$ and $\partial \psi / \partial x = 0$. ψ is therefore constant along all the edges, and we can put $\psi = 0$ for $y = 0, 1$ and $x = 0, L/d$.

The boundary conditions are then

$$\text{for } y=0 \quad \psi=0 ; \frac{\partial \psi}{\partial y}=0 ; \quad \theta=1$$

$$\text{for } y=1 \quad \psi=0 ; \frac{\partial \psi}{\partial y}=0 ; \quad \theta=0$$

$$\text{for } x=0, L/d \quad \psi=0 ; \frac{\partial \psi}{\partial x}=0 ; \quad \theta=1-y \quad \text{or} \quad \frac{\partial \theta}{\partial x}=0 \quad (2.49)$$

The system of equations shows that it is sufficient to know Ra , Pr and L/d for the determination of the ψ and θ fields.

For the solution of the above equations an iterative procedure must be applied, with suitable initial values of the ψ and θ fields. It is often necessary to increase the value of Ra in steps, the previous solution being used as the start field.

The heat flux over the whole space is

$$P = \alpha(T_H - T_C)A$$

Next to the solid surface, where only conduction takes place, we have

$$P = -\lambda \left(\frac{\partial T}{\partial y_1} \right)_{y_1=0} \cdot A$$

We thus have

$$\alpha(T_H - T_C)A = -\lambda \left(\frac{\partial T}{\partial y_1} \right)_{y_1=0} \cdot A$$

$$\frac{\alpha d}{\lambda} = - \left(\frac{\partial T}{\partial y_1} \right)_{y_1=0} \cdot \frac{d}{T_H - T_C}$$

where $Nu = \frac{\alpha d}{\lambda}$ according to equation (2.28)

Using nondimensional variables, we have

$$\frac{\alpha d}{\lambda} = - \left(\frac{\partial \theta}{\partial y} \right)_{y=0} \quad (2.50)$$

Equation (2.50) has been used for determination of the Nusselt number at a point along the vertical boundary surfaces.

In order to obtain a mean value of the Nusselt number ($\overline{Nu}$) along the vertical edge, we use

$$\overline{Nu} = \frac{d}{L} \int_0^{L/d} \frac{\alpha d}{\lambda} dx = - \frac{d}{L} \int_0^{L/d} \left(\frac{\partial \theta}{\partial y} \right)_{y=0} dx \quad (2.51)$$

In the general case, Nu is a function of Pr , Ra and L/d . For practical calculation of an air space, we use the concept thermal resistance (m)

$$m = \frac{A(T_H - T_C)}{P} = \frac{d}{\lambda Nu} \quad (2.52)$$

A temperature difference across two horizontal sides will give rise to convective motion only above a certain critical value of the Rayleigh number. It is evident from the equations for a vertical air space that even a small temperature difference, i.e. a low value of Ra, will initiate convective motion.

TAB.2.3.1 Thermal data for dry air at 760 mm Hg according to VDI-Wärmeatlas (1953) and Eckert (1972).

ϑ °C	ρ kg/m ³	c_p KJ/kgK	ν m ² /s	λ W/mK	a m ² /s	Pr	$\frac{c_{fluid}}{\rho c_p \lambda T_m}$
-50	1.534	1.005	$9.55 \cdot 10^{-6}$	0.0201	$1.32 \cdot 10^{-5}$	0.725	$0.348 \cdot 10^9$
0	1.2930	1.005	$13.30 \cdot 10^{-6}$	0.0241	$1.87 \cdot 10^{-5}$	0.715	$0.144 \cdot 10^9$
20	1.2045	1.005	$15.11 \cdot 10^{-6}$	0.0257	$2.12 \cdot 10^{-5}$	0.713	$0.104 \cdot 10^9$
40	1.1267	1.005	$16.97 \cdot 10^{-6}$	0.0273	$2.39 \cdot 10^{-5}$	0.711	$0.771 \cdot 10^8$
60	1.0595	1.009	$18.90 \cdot 10^{-6}$	0.0288	$2.67 \cdot 10^{-5}$	0.709	$0.584 \cdot 10^8$

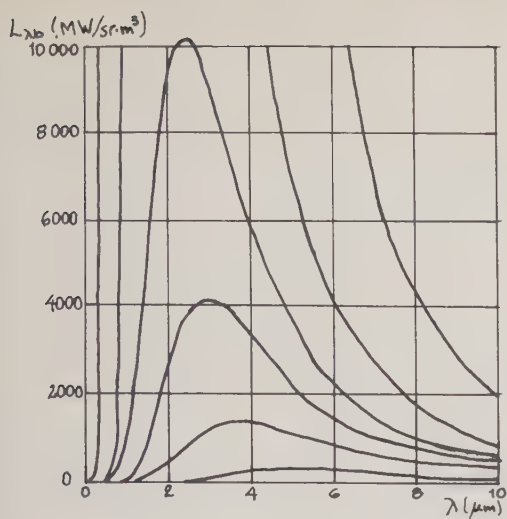


FIG.2.1.1

Spectral distribution intensity of radiance in a given direction for a black body.

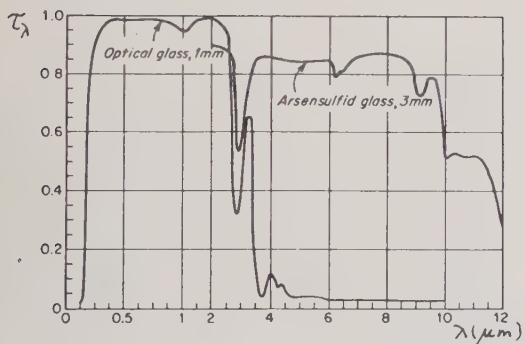


FIG.2.1.2

Transmittance through glass as a function of wavelength. Eckert (1972).

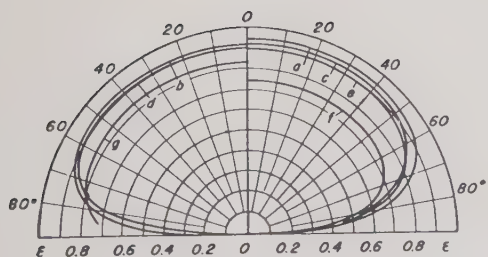


FIG.2.1.3

Emissivity in different directions. a) Wet ice, b) wood, c) paper, e) clay, f) copper oxide, g) aluminium oxide. Eckert (1972).

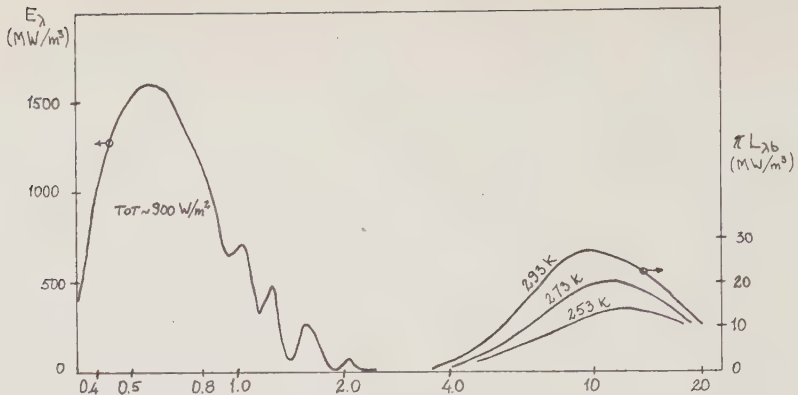


FIG.2.1.4 Spectral distribution intensity of irradiance (E_λ) for solar radiation at the earth's surface (scale at left) Spectral distribution intensity of radiance over a hemisphere for a black body at 253 K, 273 K and 293 K (scale at right), the radiant exitance M_b being 232, 315 and 418 W/m^2 respectively.

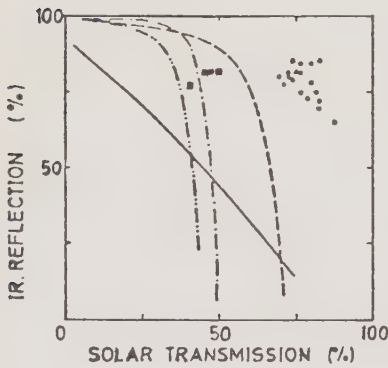


FIG.2.1.5

Infrared reflection as a function of solar transmission for different selective layers on 1 mm glass, according to Karlsson et al (1983).
 — TiN ---- Cu -.-.- Ag Au
 ■ $\text{SnO}_2(\text{NH}_4\text{F})$ ■ $\text{SnO}_2(\text{Sb}_2\text{O}_3)$

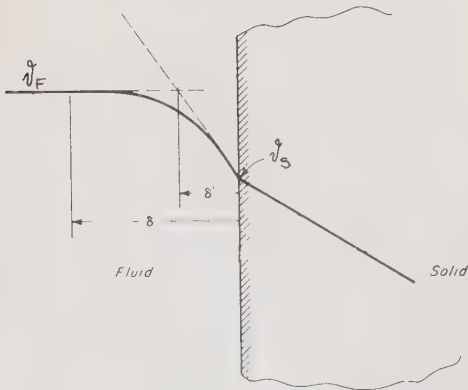


FIG.2.2.1

Convective heat transfer and conduction at a surface.

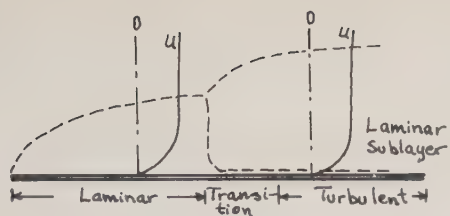


FIG.2.3.1 Velocity profile for laminar and turbulent boundary layer.

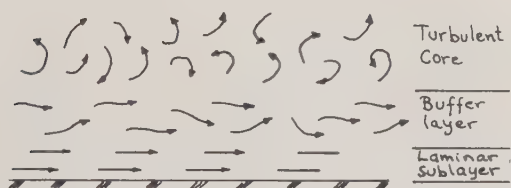


FIG.2.3.2 Structure of turbulent flow near a solid body.

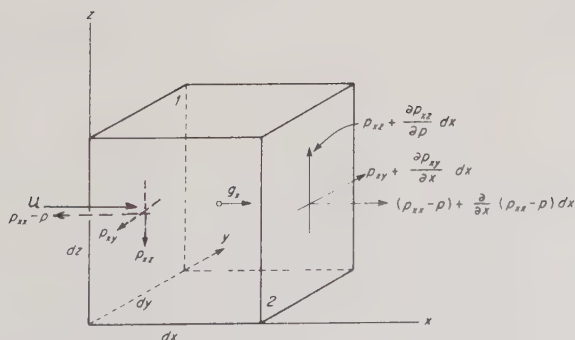


FIG.2.3.3 Stresses on a control volume.

3 NUMERICAL CALCULATIONS OF CONVECTION

For the determination of natural convection in vertical air spaces it is necessary to construct a mathematical model which describes the fundamental equations, the continuity, momentum and energy equation. There are no analytical solutions available for this system of equations, and a finite difference procedure must be resorted to. Particular study has been made of two computer programs, one by de Vahl Davis (1967) and the other by Thomas (1970). These two programs have also been documented by Efring (1974), (1975).

3.1 DESCRIPTION OF THE METHOD

In this work it has been decided to use mainly Thomas: Finite difference computation of heat transfer by natural convection. Dissertation of New South Wales (1970), for calculation of the convection process. The computer program which is valid for non-steady natural convection refers to fluids enclosed by two vertical planes and by two planes perpendicular to the vertical axis. The boundary conditions in these calculations are determined for a uniform temperature of T_H and T_C on the warm and cold side respectively. Along the horizontal edges the temperature may be supposed to have a linear distribution between the cold and warm sides or conditions may be supposed to be adiabatic, with the edges being completely isolated from the space. Variations in density which are caused by variations in the local temperature are ignored everywhere except in the term in the system of equations which describes the magnitude of the buoyant force. The other physical properties are independent of temperature. The fluid is assumed to be incompressible. It is evident from the previous chapters that the problem is characterized by three nondimensional parameters, the Rayleigh number (Ra), Prandtl number (Pr) and the aspect ratio (height/width). In solving equations (2.47) and (2.48), an auxiliary variable, the vorticity ξ , is introduced which is defined as

$$\xi = \nabla^2 \psi \quad (3.1)$$

By substituting this expression and equation (2.40) into equation (2.47), we obtain the vorticity transport equation

$$\nabla^2 \xi = Ra \frac{\partial \theta}{\partial y} + \frac{d}{aPr} \left[u \frac{\partial \xi}{\partial x} + v \frac{\partial \xi}{\partial y} \right] \quad (3.2)$$

The above equations, with their boundary conditions, are however far too complicated to be amenable to analytical solution.

The differential equations must be solved by a numerical procedure. A mesh is therefore superimposed on the space, with its axes parallel to the axes of coordinates. The outermost rows and columns coincide with the boundary surfaces. The conditions along the edges are known and are expressed by means of the boundary conditions, and a solution for the system of equations must therefore be sought for the points inside the space. The differential equations (3.2), (3.1) and (2.48) are expressed by finite differences, with the internal mesh points as the unknowns in the system of equations.

The calculation procedure may be described briefly as follows: initial values are guessed for the variables. Using these, the right hand side of equation (2.48) can be determined, yielding new values of θ . From equations (3.2) and (3.1), new values of ξ and ψ are calculated with the aid of the new value of θ and the old value of ψ .

The ADI (Alternating Direction Implicit) method has been used to solve the system of equations. Since the equations are not independent of one another and since, according to the calculation procedure, old and new values are used alternately, Thomas improved the procedure. The mean value of the old and new values of the variables is calculated and used in the final calculation. This also improves stability.

Iteration continues until the solution converges, i.e. the change between two iterations is sufficiently small. In selecting a non-dimensional time increment Δt and the number of mesh points, a compromise must be made between small values (minimum errors, long computation periods) and large values (short computation

periods but instability problems). It is possible to use cyclically varying time increments instead of constant increments in order to shorten the computation period. If a time increment is too large, an instability is introduced. If this solution is not too different from the true solution, the instability can be damped out by subsequent small time increments. The cyclical time increments generally used are 0.0005, 0.0010, 0.0025 and 0.0060.

The mesh has generally been 31×17 . For high Rayleigh numbers or large aspect ratios instability problems arise. In order to remedy this, the time increment has been reduced right down to 0.0001 and the mesh has been refined in the vertical direction, 61×17 , 121×17 and 181×17 being used. Where Ra increases steeply, an incremental procedure has generally been used, Ra being increased in increments and the previous solution used as the input data.

Computations have been carried out with values of the Rayleigh number ranging from 1000 to 400000. The aspect ratio (height/width) has been varied between 1 and 120. In order to simulate different gases in the space, the Prandtl number has been varied from 0.69 to 0.89. Boundary conditions for the horizontal edge have been either adiabatic (insulated edge) or based on a linear temperature distribution. The vertical planes have been given the nondimensional temperature 0 and 1.

3.2 CHARACTERIZATION OF THE CONVECTION REGIMES

Eckert, Carlsson (1961) studied flow in a space and described temperature distribution quantitatively with the aid of an interferogram. They classified flow into three regimes, the conduction regime, transition regime and boundary layer regime. In FIG.3.2.1 this classification has been supplemented by studies made by Yin, Wung, Chen (1978).

This classification is based on the temperature field and the temperature gradient in the horizontal direction in the middle of the space. In the conduction regime, heat transfer in the central parts of the space takes place mainly by conduction. The iso-

therms are almost vertical and the temperature gradient β_h in the horizontal direction is approximately equal to the solution for only conduction, i.e. -1 (nondimensional). As convection increases, the temperature curves begin to deflect, first in the upper and lower regions and then right in the centre of the space. The temperature gradient β_h increases to a value between -1 and 0. This region is called transition regime. As convection further increases, the temperature curves assume an increasingly rounded shape. When the temperature gradient is zero, there is no heat transfer at the centre of the space by conduction. This regime is the boundary layer regime. The temperature gradient at the centre of the space is a function of the Rayleigh number divided by the aspect ratio ($Ra/(L/d)$), as illustrated in FIG. 3.2.2. It is seen from the figure that the temperature gradient β_h has a maximum value greater than zero. This means that heat flux at the midpoint of the space is from the cold towards the warm side, i.e. its direction is opposed to that of the resultant flux. For larger values of Ra , the value of β_h decreases and approaches $\beta_h=0$.

The Prandtl number has little influence on the streamlines. In most calculations the Prandtl number for air ($Pr=0.715$) is used. If the Prandtl number varies a lot, it is not sufficient to characterize flow only by the Grashof number, but the Prandtl number must also be known. Since $Ra=GrPr$, the Rayleigh number has been used as the governing parameter for classification of flow.

As Ra increases, flow changes from a unicellular flow to one containing an increasing number of rolls (secondary flow), the ratio of height to width in the central rolls becoming increasingly small. It is difficult to determine at which critical value of Ra these secondary rolls are formed, since they are indistinct and difficult to detect. Elder (1965) estimated that the secondary flow occurs at about $Ra=3 \cdot 10^5$ for $L/d \approx 10-20$.

At a value of Ra near 10^6 , there is a sudden change in the weak shear layers between rolls. The layers become thicker and a new type of flow begins in the form of small tertiary rolls in the shear layers between the secondary rolls (tertiary flow). These tertiary rolls move in an anticlockwise direction, while the

secondary and primary rolls circulate in a clockwise direction. By simulating cases with high values of Ra , de Vahl Davis, Malinson (1975) carried out numerical calculations for $L/d=10$ (FIG.3.2.3, 3.2.4 and 3.2.5). At $Ra=5 \cdot 10^5$ secondary flows occur near the centre and become more vigorous as Ra increases. For the central rolls, the aspect ratio (height/width) is 2.45 and 2.00 for $Ra=9.4 \cdot 10^5$ and $Ra=3.3 \cdot 10^6$ respectively. Tertiary flow occurs at $Ra=9.4 \cdot 10^5$. For $L/d=20$ and $Ra=9.4 \cdot 10^5$, the flow comprises five secondary rolls (three for $L/d=10$) and four weak tertiary rolls.

Below the critical Rayleigh number the temperature along the vertical centre line is almost linear (primary flow). As the critical value of Ra is exceeded, the primary and secondary flows are superimposed. FIG.3.2.5 shows the way in which the temperature oscillates about a linear temperature curve.

Plateaus of constant temperature represent secondary rolls with cores of constant velocity. In these rolls the dominant heat transfer mechanism is convection, as can be seen from the oscillations of the local Nusselt number and the coincidence of the isotherms and streamlines in the central region. Where the temperature change is large, shear layers are set up between rolls. By photographing particles, Seki, Fukusako, Inaba (1978) have even studied the transition between different types of flow.

Elder (1965) also studied the transition to turbulent flow. For values of Ra above approx $8 \cdot 10^8 (Pr)^{0.5} (L/d)^{-3}$, wavelike movements begin to extend from the warm to the cold wall. At higher values of Ra when the wave amplitude is small, the phases between successive wave fronts become more and more random in character. At approximately $Ra=1.0 \cdot 10^{10} (L/d)^{-3}$, an intensive mixing process begins between the wall region and the inner region. The centre of the inner region is turbulent, and the region increases further towards the ends as Ra increases.

3.3 TEMPERATURE DISTRIBUTION

The horizontal temperature distribution for $Ra=1000, 4000, 16000, 64000, 107000$ and 140000 , $L/d=10$ and $Pr=0.715$ has been plotted in FIG.3.3.1 and 3.3.2. It is evident from the analytical solution that for pure conduction the temperature gradient is -1 , i.e. a straight line, which can be studied at $Ra=1000$. At larger values of Ra the curve increasingly deflects, a boundary layer begins to form at the warm and cold surfaces. There is increasing contribution to heat flow by convection. It is seen from FIG.3.3.1 that the temperature curve is approximately horizontal in the central portions and that the boundary layer becomes increasingly distinct. The thickness of this layer increases towards the top on the warm side and towards the bottom on the cold side. This can be designated the boundary layer regime. It can also be seen at high values of Ra that the temperature gradient θ_h in the centre is slightly positive, i.e. thermal conduction in the central portions has a direction opposite to that of the general heat flux. The way in which the temperature gradient θ_h varies with the Rayleigh number and the aspect ratio is shown in FIG.3.3.11.

FIG.3.3.3 and 3.3.4 illustrate the horizontal temperature distribution for aspect ratios of 20 and 40 respectively. According to FIG.3.2.1 flow conditions change if L/d is altered. This can be seen for $Ra=16000$ which for $L/d=40$ has an almost linear temperature curve, i.e. the conduction regime. As L/d decreases, the curve increasingly deflects, and for $L/d=10$ it has an almost horizontal portion, i.e. the boundary layer regime.

The temperature along vertical lines for different values of y_1/d is plotted in FIG.3.3.5-3.3.6. It can be seen in FIG.3.3.5 that temperature distribution is linear over the greatest part of the height, it is only the regions nearest the edges which are disturbed. As the value of Ra rises the effects move inwards from the edge zones, and the curve assumes a slight S shape. In the boundary layer regime (FIG.3.3.6) the curve is approximately linear in proportion to the vertical distance from the horizontal edges.

It is seen from FIG.3.3.7-3.3.9 that for high values of Ra the temperature is a linear function of height, and that the temperature gradient Δ_h is approximately zero (boundary layer regime). As the value of L/d increases with Ra remaining the same, temperature ceases to be a function of vertical distance in the centre of the space (x_1/L) and the temperature gradient approaches the value -1 (conduction regime), FIG.3.3.11. By increasing the aspect ratio at a constant value of Ra , the case can thus be transformed from the boundary layer regime, via the transition regime, to the conduction regime.

In FIG.3.3.10 all computed cases have been classified into regimes on the basis of the temperature gradient (Δ_h) at the centre of the space. Compared with data by Yin, Wung, Chen (1978), the lines between the regimes have the same inclination. The boundary between the conduction and transition regimes is at about the same place, while the boundary between the transition and boundary layer regimes has been displaced laterally by a relatively large amount. The results of this investigation point to the transition regime having a much narrower band than in previous investigations. The reason for this discrepancy is that the investigations referred to have been experimental ones while this investigation contains theoretically calculated values (see also FIG.6.2.18).

3.4 VELOCITY AND FLOW DISTRIBUTION

FIG.3.4.1 sets out the vertical velocity at $x_1/L=0.50$ for $Ra=1000, 4000, 16000, 64000, 107000$ and 140000 . Apart from the direction of flow, the profiles are symmetrical about the centre line. As Ra increases, the maximum velocity rises and the thickness of the boundary layer decreases. For $Ra=1000, 4000$ and 16000 , the curves have a point of inflexion at $y_1/d=0.5$. At higher values of Ra the curves have three points of inflexion, with one situated at $y_1/d=0.5$. The distance between the other two increases with increasing values of Ra .

Elder (1965) gives a general discussion of the mechanisms needed to create convective motion. A temperature gradient in the verti-

cal direction causes the fluid to accelerate at the bottom since the buoyant forces exceeded the viscous forces. The buoyant forces diminish until there is a balance with viscous forces at half the height. After this the fluid is retarded. Outside the layer near the wall, there is a horizontal inward flow whose velocity is negative in the bottom half and positive in the top half. This corresponds to inflow of fluid into the expanding layer and to one out of this when growth of this has attained a maximum. At half the height, the rate of inflow is approximately zero for reasons of continuity and symmetry.

Appendix 1 to Jonsson (1985) presents flow patterns and isotherms. For some cases, the vorticity ξ and the velocities u , v , and $\sqrt{u^2+v^2}$ are also given. These illustrate the way in which flow changes from unicellular to multicellular. For $L/d=10$ there is a change at $Ra \approx 10^5$. As Ra increases, convection increases all the time, and the maximum value of the stream function which is located at the midpoint of the convection roll also increases. FIG.3.4.2 shows the way in which the maximum flow increases as the value of Ra rises. In addition, the unicellular flow is transformed into multicellular flow.

Calculations have shown that under certain conditions flow becomes unstable and a multicellular convection pattern develops. This occurs for $L/d=20$ at values of Ra between approx 12000 and approx 40000, and for $L/d=40$ at values of Ra between approx 16000 and approx 40000. In addition, flow returns to being unicellular, and this is maintained until multicellular behaviour begins in the boundary layer regime for $L/d=20$ at $Ra=120000$. The multicellular pattern at low values of the Rayleigh number represents a transition from the conduction to the transition regime. No similar behaviour has been detected for values of L/d other than $L/d=20$ and $L/d=40$.

These instability problems for elongated spaces with large values of L/d have also been demonstrated by experiments and calculations, Korpela, Yee Lee, Drummond (1980). In attempts to optimize the space width (d) for constant height (L), it was found that in the conduction regime heat transfer decreased as the width of space increased. As flow changes from the conduction to

the transition regime, rate of flow rises, and for certain values of L/d there is a change to multicellular behaviour. In the boundary layer regime heat flow is comparatively independent of the space width. If a correlation is carried out of the form

$$Nu = A Ra^B (L/d)^C \quad (3.3)$$

the heat flow rate will increase as the width of space increases, if $3B-C-1 > 0$. Comparisons with formulae by Thomas (1970), see TAB.3.5.1, show that in the conduction region $3B-C-1 = -0.645$, i.e. < 0 , and the heat flow rate decreases as the width of spaces is increased. For the transition and boundary layer regimes, $3B-C-1 = 0.128$ and 0.012 respectively, i.e. > 0 . In the transition regime there is thus a slight rise in heat flow rate as the width of space increases, and in the boundary layer regime the heat flow remains relatively constant as the width of the space increases. The width of space would thus have an optimum value with respect to heat flow rate at the point of transition between the conduction and transition regimes. It is here that for certain values of L/d a multicellular convection pattern is set up.

3.5 HEAT TRANSFER

The Nusselt number has been computed for the cases previously calculated, and is illustrated in FIG.3.5.1-3.5.3. For increasing values of the Rayleigh number the heat transfer rate increases, i.e. the value of the Nusselt number rises. However, the total heat transfer is greatly dependent on x_1/L , i.e. the distance from the upper edge. Convection begins at the bottom corner on the warm side and at the top corner on the cold side. The surface coefficients of heat transfer are therefore higher at these points, and they are called starting corners. The surface coefficient of heat transfer then decreases with height on the warm side and is a minimum at the top corner which is therefore called departure corner.

In some cases (FIG.3.5.2 and 3.5.3) pronounced peaks develop due to the formation of multicellular flow. This can also be compared to the temperature gradient in the vertical direction, see FIG.

3.3.8 and 3.3.9. For e.g. $Ra=200000$ and $L/d=20$, a small roll is formed at the bottom of the space.

Heat transfer in the conduction regime is characterized by the fact that conduction takes place in the central portions of the air space, i.e. $Nu=1$. The local heat transfer rates at the corners are however not the same. The bottom corner on the warm side and the top corner on the cold side have a surface coefficient of heat transfer higher than the average (starting corners). The other corners have a lower than average surface coefficient of heat transfer (departure corners). So long as the end effects do not influence the central regions, the local surface coefficients of heat transfer at the corners are independent of the height L . The local Nusselt number is then a function of Gr_x and x/d , i.e. $Nu=f(Gr_x, x/d)$. Eckert, Carlsson (1961) fitted their experimental data to this relationship. They found that for small values of the Rayleigh number the influence of the starting corner is confined to a distance from the corner equal to 1-2 times the width of the space. As Ra increases, the starting corner extends towards the centre, until at $Ra=2000$, $L/d=10$ the whole space is affected.

When a vertical space is divided up into several smaller ones by means of horizontal walls (L/d decreases), the rate of heat flow increases. As convection begins in the bottom warm corner (or the top cold corner), the starting corners, large differences in temperature develop between the fluid and the vertical surface. The fluid accelerates upwards and in this way the rate of heat flow increases at the starting corner. The Nusselt number has a more or less pronounced peak at the starting corner. The magnitude of this depends on Ra and the height/width ratio. In the upper part of the space the local Nusselt number is approximately constant within the conduction regime (see 3.5.1-3.5.3). If a space is divided up into a number of subspaces, convection (the convection rolls) will also be divided up into number of cells. In each cell, convection begins at the bottom warm corner, and the local Nusselt number will have a peak here. In this way a number of peaks will be superimposed on the original distribution of the Nusselt number. The average value of the Nusselt number will be higher, and the heat flow rate will therefore increase. This

state of affairs may be compared to the formation of a number of rolls in a space, in this case also the rate of heat flow rises. This will also be discussed in section 3.7. So long as the end effects do not influence the central regions, the local surface coefficients of heat transfer are independent of height (FIG. 3.5.1 Ra=1000).

Meyer, Mitchell, El-Wakil (1979) have studied convection between vertical parallel panels separated by horizontal spacers. If, from $L/d=20-40$, the spacing of the horizontal separators is increased, Nu will have a maximum at $L/d=1-2$. When the value of L/d drops below 1, the viscous forces (shear forces) against the cell walls will assume an increasing influence, and the motion of the fluid will be retarded and the convective heat flow rate diminish.

As the convection conditions change when the aspect ratio is altered, the heat transfer rate at the surfaces will be affected. If a space is made more elongated (L/d increased at a constant value of Ra), flow may change from the boundary layer regime to the conduction regime. At higher values of L/d the Nusselt number is almost constant and approximately equal to 1, while at lower values there is a pronounced peak at the starting corner and the average values over the entire height are higher.

Generally, the average Nusselt number may be said to be a function of the Rayleigh number (Ra) the Prandtl number (Pr), the aspect ratio (L/d) and the ratio of the width of window to the width of air space (b/d). Under two-dimensional conditions the last term vanishes.

$$Nu_d = \frac{\alpha_c^d}{\lambda} = f(Ra_d, Pr, \frac{L}{d}, \frac{b}{d}) \quad (3.4)$$

For two-dimensional flow the assumption usually made is

$$Nu_d = A Ra_d^B Pr^C (L/d)^D \quad (3.5)$$

This expression is transformed into the relationship

$$\log Nu_d = \log A + B \log Ra_d + C \log Pr + D \log(L/d)$$

for correlation of the available readings according to the method of least squares.

A number of studies have been summarized in FIG.3.5.4 (Anderlind 1972) and in TAB.3.5.1 which sets out theoretical (T), experimental (E) or numerical (N) calculations. Seki, Fukusako, Inaba (1978) used the height (L) of the space instead of its width (d) as the characteristic length.

The exponent D of the height/width ratio L/d is given by Jakob (1946) as -0.111, by Eckert, Carlsson (1961) as -0.1 and by Yin, Wung, Chen (1978) as -0.131, while Newell, Schmidt (1970), MacGregor, Emery (1969) and Thomas (1970) give -0.256, -0.30 and -0.246 respectively. Objections have been made to some earlier investigations on the grounds that the height L was small and that excessively simplified assumptions had been used for the theoretical model. In later investigations there is a difference between numerical and experimental methods, the exponent D being approximately -0.25 and -0.13 respectively. Conversion of the values according to Seki, Fukusako, Inaba (1978) to Nu_d puts the exponent D at -0.36, -0.1 and -0.1 respectively.

The majority of authors state that the Prandtl number exerts little influence, the exponent is small. According to Thomas (1970), Seki, Fukusako, Inaba (1978) and Dropkin, Somerscale (1965), the value of C is between 0.006 and 0.096.

The exponent of the Rayleigh number has been based by some authors on the character of the flow, 0.25 for laminar conditions and 0.33 for turbulent conditions. Others, by minimization according to the method of least squares, obtained a value between 0.101 and 0.37. In general the value of the exponent is between 0.26 and 0.30 for $L/d > 5$.

Since there is so much variation in the exponents, the constants cannot be compared directly. An increase (decrease) in the value

of an exponent affects the values of the other exponents.

A program drawn up by Thomas (1970) has been used for calculation of the convection process. Values calculated according to his formulae in TAB.3.5.1 should therefore be approximately the same as values in FIG.3.7.11-3.7.12. When they are compared, the difference for $L/d=10, 20$ and 40 is less than about 5%. For large values of L/d , the equation does not hold for the conduction regime. If this equation is nevertheless used, the values of Nu are too small, less than 1. For large values of L/d the exponent of L/d diminishes. In this way the variation in the Nusselt number is very slight, in FIG.3.7.12 Nu is between 1 and 1.1.

3.6 DIFFERENT GASES

When gases other than air are used in the space, the character of conductive and convective heat transfer changes. For practical reasons, the gas used

- 1 shall not be noxious
- 2 shall be resistant to UV radiation and unaffected by changes in temperature
- 3 shall be chemically inert in relation to the sealing materials used
- 4 shall not condense at low temperature ($<-35^{\circ}\text{C}$)
- 5 its permeability through the edge seal shall be zero for all practical purposes

Gases which satisfy these conditions are air, argon (Ar), sulphur hexafluoride (SF_6) and krypton (Kr).

In a plate apparatus, Gläser (1977) investigated the gases air, argon and sulphur hexafluoride and mixtures of these in spaces of widths ranging between 6 and 20 mm. A number of authors, e.g. Mahdjuri (1977), Selkowitz (1979), Krings, Olink (1957), Owens, Barnett (1974), have studied the influence of different gases. Freyholdt (1980) has studied sealed units with the air space at low pressure. Selkowitz (1979) states that when pressure is reduced below one atmosphere, heat transfer by convection is reduced

until only conduction takes place, which is proportional to the width of space and the thermal conductivity of the gas. This state of affairs continues until the mean free path of the molecules is larger than the dimension of the space, at which point conduction is also reduced. This occurs at pressures less than about 1-10 Pa.

In order to find how different gases influence conditions in a space, calculations have been made for the gases air, argon (Ar), carbon dioxide (CO_2), sulphur dioxide (SO_2) and sulphur hexafluoride (SF_6). The height/width ratio has been put at 100, 75 and 50. This means that for a sealed unit of 0.6 m height the space between panes is 0.006, 0.008 and 0.012 m. The difference in temperature has been assumed to be 20 K, and the heat flow rate at the horizontal edges has been put at zero. The physical properties are set out in TAB.3.6.1. For small space widths, convection is small for all gases (FIG.3.6.2). If the width of space is increased, convection also increases, but in the case of air and argon less than for the other gases (FIG.3.6.3). It is evident from FIG.3.6.1 that the vertical gradient for air, argon and carbon dioxide is constant over the greater portion of the space, which is typical for the conduction regime. In the case of the other gases, SO_2 and SF_6 , convection increases, and flow may be considered to be characteristic of the transition regime or boundary layer regime. When a choice has to be made between the three gases commonly used, air, argon and sulphur hexafluoride, it is convection that determines the choice. Since air and argon have approximately the same convective properties, argon will at all times produce lower heat transfer. The convection of SF_6 is small only in the case of narrow spaces. FIG.3.6.4 illustrates the thermal resistance of the window with $\alpha_R = 3.8 \text{ W/m}^2\text{K}$. A comparison with Gläser (1977) shows that the general appearance of the curves is in good agreement. There is comparatively little difference in the values of thermal resistance.

3.7 TEMPERATURE ALONG THE VERTICAL EDGES

Calculations and measurements of the convective heat transfer in a vertical space under the influence of natural convection have previously generally been carried out with the temperature along the vertical edges maintained at a constant value. For a window in a building, the boundary conditions on the outside, in a strong wind, may approach constant temperature conditions. On the inside, conditions may vary depending e.g. on geometrical configuration and the ventilation and heating systems. However, the motive forces are smaller than on the outside, and air movements on the inside of the window are small. Variations in temperature on the inside of the window will be larger. During measurements in the laboratory, the author endeavoured to simulate the conditions to which a window is exposed in a building. In the cold room there are fans which create a horizontal air flow perpendicular to the window. In this way the vertical temperature gradient is small. On the warm side (in the hot box), movement of air takes place only due to natural convection. In this way the temperature variation in the vertical direction is comparatively large.

Convection has been computed for test windows P1-P4. Values measured for test windows according to APP.6.2.6-6.2.9 have been used as input data. The calculated distribution of the Nusselt number along the warm side is illustrated in FIG.3.7.1-3.7.5. When these values are compared with the measured heat flows, agreement is not very satisfactory. Although certain trends are the same for the calculated and measured flows, there are a number of discrepancies which are difficult to explain. FIG.3.7.5 shows the way in which the Nusselt number varies along the inside and along the intermediate pane in test window P2 which has two air spaces. The curve for the pane between the air spaces is approximately symmetrical, with pronounced peaks at the top and bottom corners.

Generally speaking, the choice of input data such as the temperature distribution along the edges may considerably influence flow configuration and thus flow along the edges. In order to study the way in which boundary conditions influence the convection

process, the temperatures measured on test windows P1-P4 have been used. For a certain measurement, the temperature distribution along the edges will relate to a definite value of L/d and Ra . In order that comparisons may be made with the more general cases of $L/d=10, 20$ etc, an endeavour has been made to fit the measured values to these set values of L/d . The mean temperature for all cases has been put at 1 or 0 along the vertical edges (see APP.6.2.6-6.2.10). In calculations an endeavour has been made to make the value of L/d as nearly as possible equal to 10, 20, 40, 80 or 120. For instance, the temperature distribution along the vertical edges for case P2-40 with $L/d=21$ and case P3-30 with $L/d=18.7$ has been used for calculations with $L/d=20$. In this way the boundary conditions will not exactly fit the calculated cases, but will be near the "true" values. Generally speaking, it is practically impossible to specify a "true" value of the temperature distribution, since it is affected by so many factors. The temperature along the edges, or the way in which heat transfer takes place at the inside and outside faces in conjunction with natural or forced convection, or the mean ambient radiant temperature, may for instance be affected by the frame/casement construction. On the other hand, the temperature at the same horizontal level is approximately constant, if it varies by $\pm 0.5^\circ\text{C}$, this gives rise to $\pm 2-3\%$ at a temperature difference of 20°C .

In FIG.3.7.6-3.7.7 case P4-20 has been fitted to the calculations with $L/d=10$. When a comparison is made with constant temperatures along the edges, it is seen that there is a slight change in the vertical temperature distribution (FIG.3.7.6). In addition, the distribution of the local Nusselt number is also a little different. It is particularly at the top of the space that the Nusselt number increases (FIG.3.7.7), and this also increases the average value of the Nusselt number. In FIG.3.7.8 and 3.7.9 some cases have been fitted to $L/d=20$ and different values of the Rayleigh number. For some values of Ra , for instance 140000, there are pronounced peaks in the distribution of the local Nusselt number. This shows that the convection pattern has changed and a flow with several rolls has developed. Generally speaking, the selected boundary conditions exert great influence on the distribution of the Nusselt number. Let us examine a case with $Ra=107000$ and

$L/d=20$ for different boundary conditions. In FIG.3.5.2 the value $\bar{Nu}=2.70$ is obtained for the temperatures 1 and 0 along the horizontal edges. In FIG.3.7.8 this rises to $\bar{Nu}=3.02$ for edges with modified temperatures. Calculations for the test window P3-30, with $Ra=108000$ and $L/d=18.7$, produce $\bar{Nu}=2.64$ as shown in FIG.3.7.3. It is also to be noted that there is a change in the distribution of the Nusselt number, from a very pronounced peak at the bottom corner to a much more uniform distribution - in the central portions the maximum deviation of Nu from the mean value is $\pm 30\%$. The appearance of the curve representing the Nusselt number is thus greatly dependent on the temperatures assumed along the edges. However, there is relatively little change in the average Nusselt number between uniform temperatures (FIG.3.5.1-3.5.3) and calculations for measured cases (FIG.3.7.1-3.7.4). On the other hand, the value of Nu may considerably increase if the measured temperatures are fitted to $L/d=10, 20$, etc. This would seem to be due to the fact that temperature distribution along the edge is not quite correct but often gives rise to a multicellular convection pattern. In turn, this causes the value of the Nusselt number to increase. It should be noted that calculations for test windows P1-P4 show unicellular behaviour in all cases, which yields a low value of the Nusselt number. If there is a lot of difference between convection patterns for different calculation assumptions (unicellular-multicellular) the difference between the values of Nu will increase.

In FIG.3.7.11 and 3.7.12 the average Nusselt number is plotted in relation to the Rayleigh number for $L/d=1, 2, 5, 10, 20, 40, 80$ and 120. The average Nusselt number is approximately the same in calculations with the input data based on measured cases and in calculations with constant temperatures along the vertical edges. If attempts are made to fit the measured cases to other values of L/d or Ra , disturbances easily occur in the convection pattern. More cells are formed, and the average Nusselt number increases. The average value of the Nusselt number is higher for adiabatic (totally insulated) horizontal edges than for horizontal edges with a linear temperature distribution. This can be explained with the aid of FIG.3.7.10.

For horizontal edges with a linear temperature distribution, the air temperature near the edges will be appreciably different from the surface temperature; the higher the Rayleigh number, the greater this deviation. This means that there are large temperature gradients along the horizontal edges, and a heat flux will pass along the upper and lower horizontal portions of the vertical air space. In the lower portion this flux moves into the air space, while in the upper portion it moves out of the air space. This flux also contributes to convection in the space. If the Nusselt number is therefore calculated only along the warm vertical surface, the vertical flow will not be included in the calculations. Compared with adiabatic conditions, horizontal edges with a linear temperature distribution will have a lower value of the Nusselt number for otherwise similar conditions. The magnitude of this difference is a function of the Rayleigh number. A comparison of the mean temperature difference at the horizontal edges (FIG.3.7.10) with the difference in the values of the Nusselt number for edges with linear temperature distribution and for adiabatic edges gives

Ra	Mean temperature difference		Difference in $\bar{Nu}$ (linear-adiabatic)
	upper edge	lower edge	
1000	0.093	0.093	0.00
4000	0.240	0.241	0.08
16000	0.369	0.372	0.14
64000	0.414	0.414	0.28
107000	0.425	0.426	0.38
140000	0.427	0.430	0.46

TAB.3.5.1 Average values of the Nusselt number. T=theoretical, E=experimental and N=numerical determination.

Reference	Case	$Nu_d (Nu_L)$	Limitations
Batchelor (1954)	T	$0.43(Gr_d)^{0.25} (\frac{L}{d})^{-0.25}$	$\frac{L}{d} < \frac{Ra_d}{500}$ air
	T	$1 + 0.00139 Ra_d (\frac{L}{d})^{-1}$	$\frac{L}{d} > \frac{Ra_d}{500}$ air
Jakob (1946)	E	$0.18(Gr_d)^{1/4} (\frac{L}{d})^{-1/9}$	$2 \cdot 10^4 < Ra_d < 2 \cdot 10^5$ $3.1 < \frac{L}{d} < 42$
	E	$0.065(Gr_d)^{1/3} \cdot (\frac{L}{d})^{-1/9}$	$2 \cdot 10^5 < Ra_d < 10^7$ $3.1 < \frac{L}{d} < 42$
De Graaf, van der Held (1952)	E	1	$Gr_d < 7 \cdot 10^3$
	E	$0.0384(Gr_d)^{0.37}$	$10^4 < Gr_d < 8 \cdot 10^4$
	E	$0.0317(Gr_d)^{0.37}$	$Gr_d > 2 \cdot 10^5$
Eckert, Carlsson (1961)	E	$1 + 0.00166(Gr_d)^{0.9} (\frac{L}{d})^{-1}$	Conduction regime
	E	$0.119(Gr_d)^{0.3} (\frac{L}{d})^{-0.1}$	Boundary layer regime $2.3 < \frac{L}{d} < 4.6$ $8.0 \cdot 10^4 < Gr_d < 20 \cdot 10^5$
Dropkin, Somerscale (1965) 1)			$0.02 < Pr < 11560$
	E	$0.049(Ra_d)^{1/3} (Pr)^{0.074}$	$5 \cdot 10^4 < Ra_d < 7.17 \cdot 10^8$
MacGregor, Emery (1969) 1)	E	$0.420(Ra_d)^{1/4} (Pr)^{0.012} \cdot (\frac{L}{d})^{-0.30}$	$10^4 < Ra_d < 10^7$ $1 < Pr < 20000$ $10 < L/d < 40$
	E	$0.046(Ra_d)^{1/3}$	$10^6 < Ra_d < 10^9$ $1 < Pr < 20$ $1 < \frac{L}{d} < 40$
Newell, Schmidt (1970)	N	$0.0547(Gr_d)^{0.397}$	$L/d=1$ $2 \cdot 10^3 < Gr_d < 7 \cdot 10^4$
	N	$0.155(Gr_d)^{0.315} (\frac{L}{d})^{-0.265}$	$2.5 < \frac{L}{d} < 20$ $4 \cdot 10^3 < Gr_d < 2 \cdot 10^4$

TAB.3.5.1 (cont)

Reference	Case	$Nu_d(Nu_L)$	Limitations
Thomas (1970)	N	$0.595(Ra_d)^{0.101}(Pr)^{0.024} \cdot (L/d)^{-0.052}$	$0 < Ra_d < 1.1 \cdot 10^4$ $0.71 < Pr < 1$ $5 < L/d < 33$ conduction regime
	N	$0.202(Ra_d)^{0.294}(Pr)^{0.096} \cdot (L/d)^{-0.246}$	$2 \cdot 10^3 < Ra_d < 3 \cdot 10^4$ $0.25 < Pr < 1$ $5 < L/d < 33$ transition regime
	N	$0.286(Ra_d)^{0.258}(Pr)^{0.006} \cdot (L/d)^{-0.238}$	$10^4 < Ra_d < 2.5 \cdot 10^5$ $0.4 < Pr < 10^4$ $5 < L/d < 20$ boundary layer regime
Seki, Fukusako, Inaba ¹⁾ (1978)	E	$0.36(Ra_L)^{0.25}(L/d)^{-0.11} \cdot Pr^{0.051}$	Laminar $10^7 < Ra_L < 4 \cdot 10^9$ $1 < Pr < 40000$ $5 < L/d < 47.5$
	E	$0.084(Ra_L)^{0.30}(Pr)^{0.051}$	Transition $4 \cdot 10^9 < Ra_L < 4 \cdot 10^{12}$ $1 < Pr < 200$ $5 < L/d < 47.5$
	E	$0.096(Ra_L)^{0.30}(Pr)^{0.051}$	Turbulent $Ra_L > 2 \cdot 10^{12}$ $1 < Pr < 4$ $5 < L/d < 47.5$
Yin, Wung, Chen (1977)	E	$0.210(Gr_d)^{0.269}(\frac{L}{d})^{-0.131}$	$1.5 \cdot 10^3 < Gr_d < 7 \cdot 10^6$
			$4.9 > L/d < 78.7$

1) Isothermal cold wall and constant heat flow through the warm wall

$$Gr_d = \frac{g\gamma(T_H - T_C)d^3}{\nu^2} \quad Gr_L = \frac{g\gamma(T_H - T_C)L^3}{\nu^2} \quad Ra = Gr \cdot \frac{\nu}{a}$$

TAB.3.6.1 Physical properties of gases at 0 °C and 760 mm Hg (1.033 at) according to Tefyma (1971).

Fluid	λ (W/mK)	ν (m ² /s)	c_p (J/kgK)	ρ (kg/m ³)	Pr	C_{fluid} ($= \frac{g \rho c_p}{\sqrt{\lambda} T_m}$)
Air	0.0241	$13.3 \cdot 10^{-6}$	$1.00 \cdot 10^3$	1.293	0.715	$0.144 \cdot 10^9$
Ar	0.016	$11.9 \cdot 10^{-6}$	$0.52 \cdot 10^3$	1.784	0.690	$0.175 \cdot 10^9$
CO ₂	0.014	$7.03 \cdot 10^{-6}$	$0.82 \cdot 10^3$	1.977	0.814	$0.592 \cdot 10^9$
SO ₂	0.0084	$3.98 \cdot 10^{-6}$	$0.64 \cdot 10^3$	2.926	0.887	$2.01 \cdot 10^9$
SF ₆	0.013	$2.21 \cdot 10^{-6}$	$0.62 \cdot 10^3$	6.602	0.691	$5.09 \cdot 10^9$

TAB.3.6.2 Calculated values of the average Nusselt number for different gases in a space of width $d=0.006$, 0.008 and 0.012 m. Values of the height/width ratio were 100, 75 and 50. Temperature difference 20 K.

Fluid			$d=0.006$		$d=0.008$		$d=0.012$	
	λ	Pr	Nu	λNu	Nu	λNu	Nu	λNu
Air	0.0241	0.715	1.00	0.0243	1.00	0.0243	1.02	0.0248
Ar	0.016	0.69	1.00	0.016	1.00	0.016	1.03	0.0165
CO ₂	0.014	0.81	1.00	0.014	1.01	0.014	1.20	0.017
SO ₂	0.0084	0.89	1.02	0.0086	1.11	0.0093	1.66	0.014
SF ₆	0.013	0.69	1.08	0.014	1.41	0.018	2.29	0.030

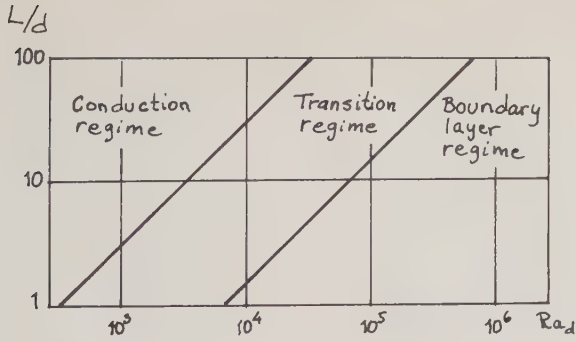


FIG.3.2.1 Classification of the three flow regimes as a function of the height/width ratio (L/d) and the Rayleigh number (Ra) for air, according to Yin, Wung, Chen (1978).

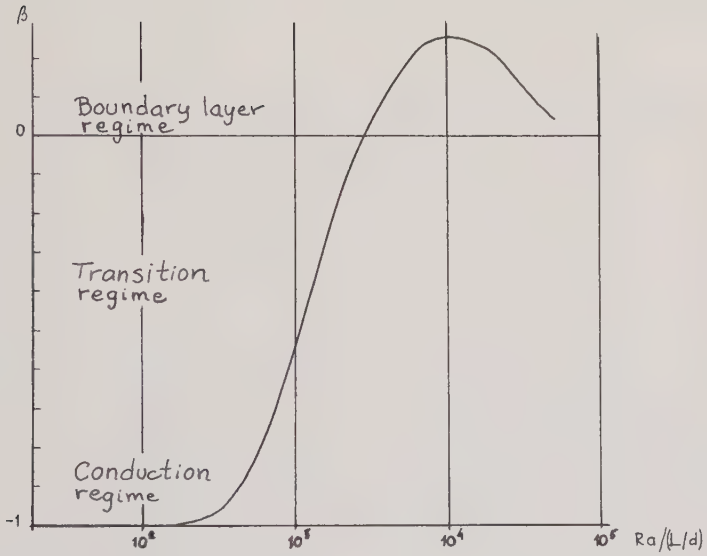


FIG.3.2.2 Temperature gradient (β_h) in the horizontal direction in the middle of the space as a function of the Rayleigh number divided by the height/width ratio ($Ra/L/d$). $Pr=1$. According to Thomas (1970).



FIG.3.2.3-3.2.4 The stream function ψ and isotherms in a space for which $L/d=10$ and $Pr=10^3$ (oil). For cases a, b, c and d according to the figure, the values of Ra are $2.4 \cdot 10^5$, $5.0 \cdot 10^5$, $9.4 \cdot 10^5$ and $3.3 \cdot 10^6$ respectively. de Vahl Davis, Mallinson (1975).

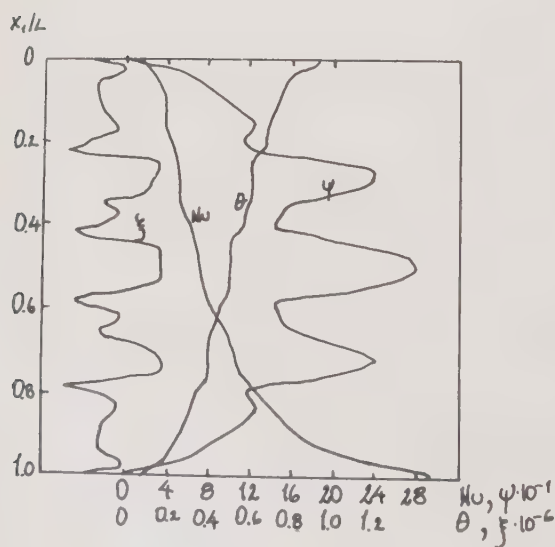


FIG.3.2.5

Distribution of temperature θ , stream function ψ and vorticity ω along the vertical centre line, and the Nusselt number for the warm wall. $Ra=3.3 \cdot 10^6$, $Pr=10^3$ (oil) and $L/d=10$. According to de Vahl Davis, Mallinson (1975).

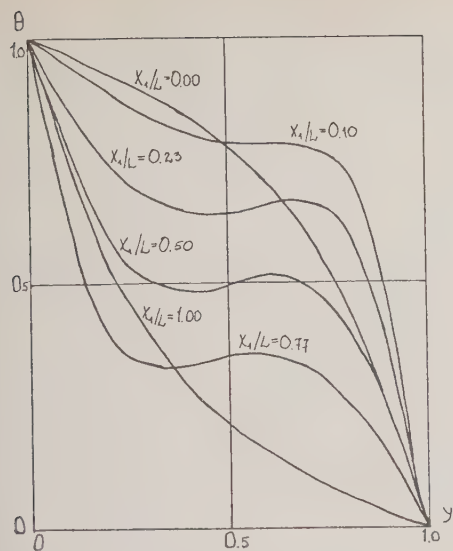


FIG. 3.3.1

Horizontal temperature distribution for $x_1/L = 0, 0.10, 0.23, 0.50, 0.77$ and 1.00 . $L/d=10$ and $Ra=64000$. Adiabatic horizontal edges.

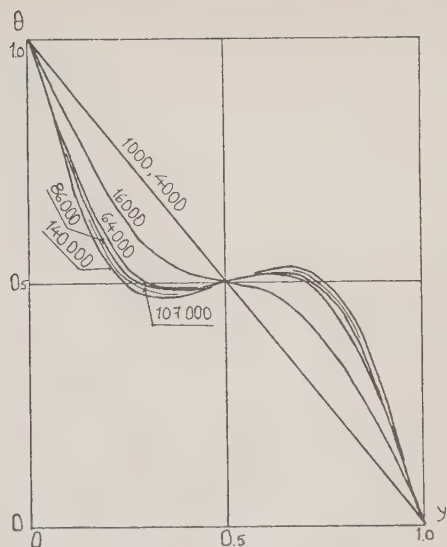


FIG. 3.3.2

Horizontal temperature distribution for $x_1/L=0.50$. $L/d=10$ and $Ra=1000, 4000, 16000, 64000, 86000, 107000$ and 140000 .

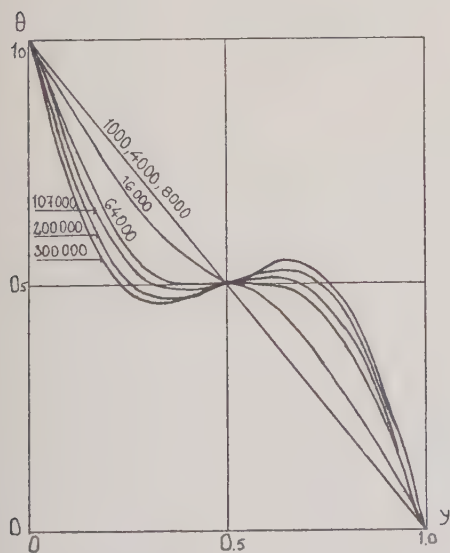


FIG. 3.3.3

Horizontal temperature distribution for $x_1/L=0.50$. $L/d=20$ and $Ra=1000, 4000, 8000, 16000, 64000, 107000, 200000$ and 300000 .

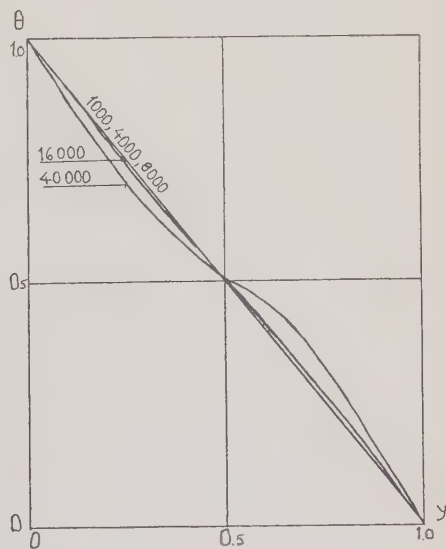


FIG. 3.3.4

Horizontal temperature distribution for $x_1/L=0.50$. $L/d=40$ and $Ra=1000, 4000, 8000, 16000$ and 40000 .

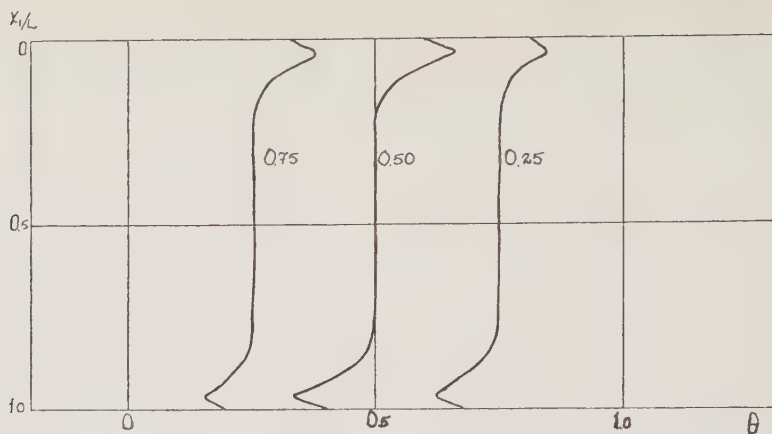


FIG.3.3.5 Vertical temperature distribution for $y_1/d=0.25$, 0.50 and 0.75 . $L/d=10$, $Pr=0.715$ and $Ra=1000$. Adiabatic horizontal edges.

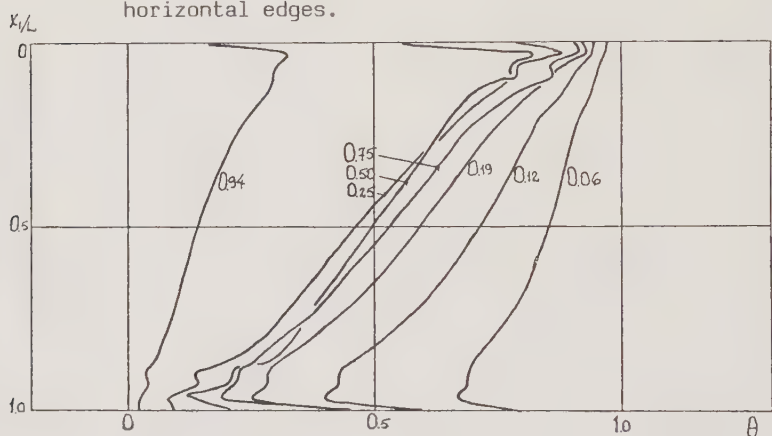


FIG.3.3.6 Vertical temperature distribution for $y_1/d=0.06$, 0.12 , 0.19 , 0.25 , 0.50 , 0.75 and 0.94 . $L/d=10$, $Pr=0.715$ and $Ra=64000$. Adiabatic horizontal edges.

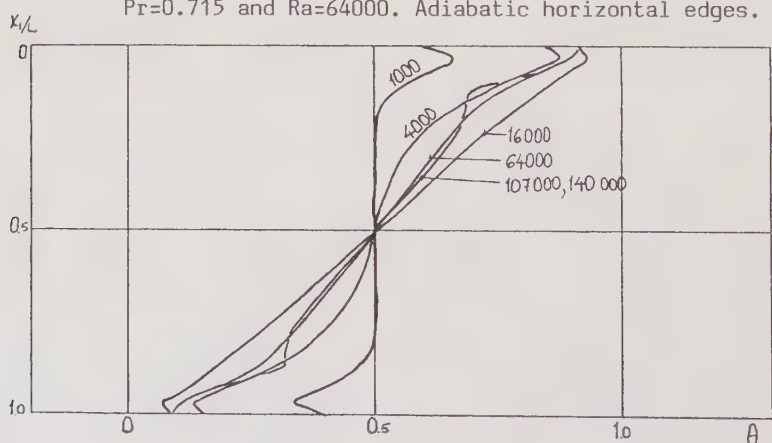


FIG.3.3.7 Vertical temperature distribution for $y_1/d=0.50$. $L/d=10$, $Ra=1000$, 4000 , 16000 , 64000 , 107000 and 140000 .

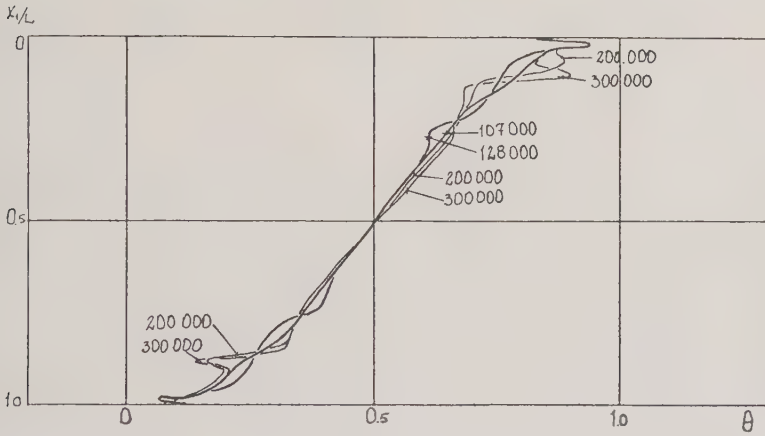
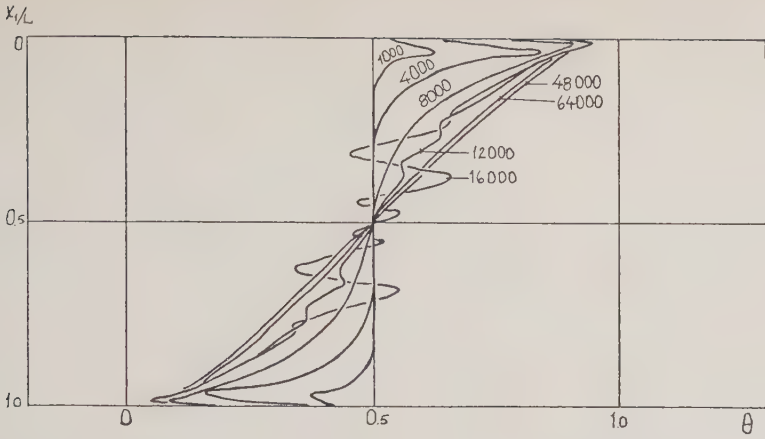


FIG.3.3.8 Vertical temperature distribution for $y_1/d=0.50$.
 $L/d=20$, $Ra=1000, 4000, 12000, 16000, 48000, 64000,$
 $107000, 128000, 200000$ and 300000 .

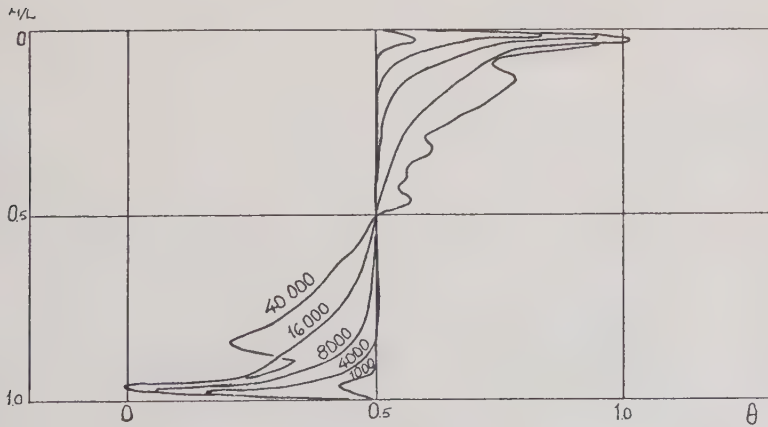


FIG.3.3.9 Vertical temperature distribution for $y_1/d=0.50$.
 $L/d=40$, $Ra=1000, 4000, 8000, 16000$ and 40000 .



FIG.3.3.10 Classification into flow regimes on the basis of the height/width ratio (L/d) and the Rayleigh number (Ra). x conduction regime, o transition regime ($-0.94 < \beta_h < 0$), $\otimes$ boundary layer regime. --- cases calculated in this report, -- according to Yin, Wung, Chen (1978) with $-1 < \beta_h < 0$ for the transition regime.

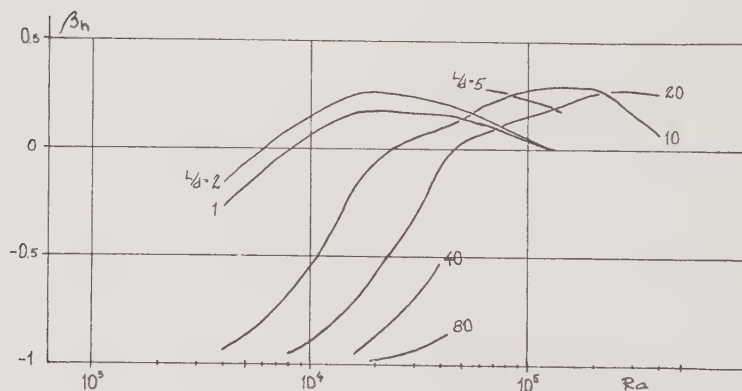


FIG.3.3.11 Temperature gradient in the horizontal direction (β_h) in the middle of the space as a function of the Rayleigh number for different height/width ratios (L/d).

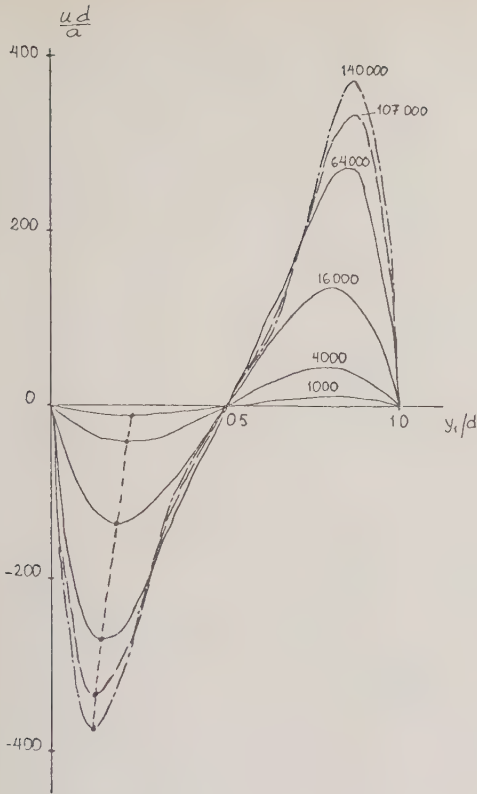


FIG.3.4.1

Vertical velocity (u) in the middle of the space for $Ra=1000, 4000, 16000, 107000$ and 140000 . $Pr=0.715$, $L/d=10$ and $x_1/L=0.50$. For air, $a=1.87 \cdot 10^{-5}$ at $0^\circ C$.

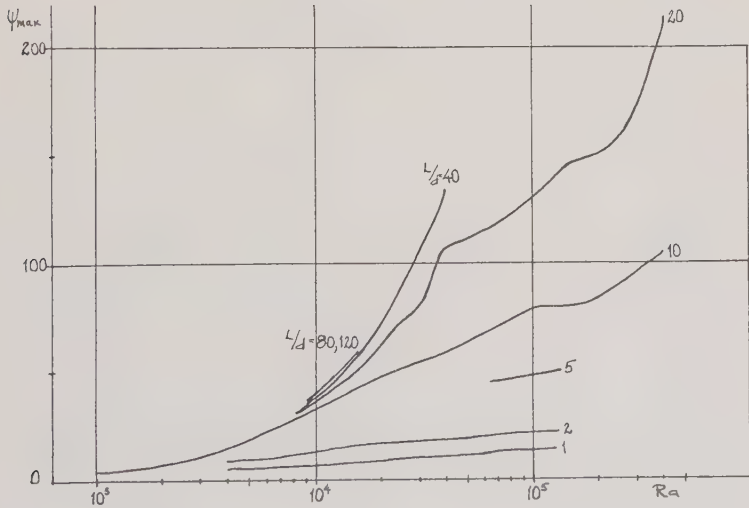


FIG.3.4.2 Maximum value of the stream function for different values of the Rayleigh number.

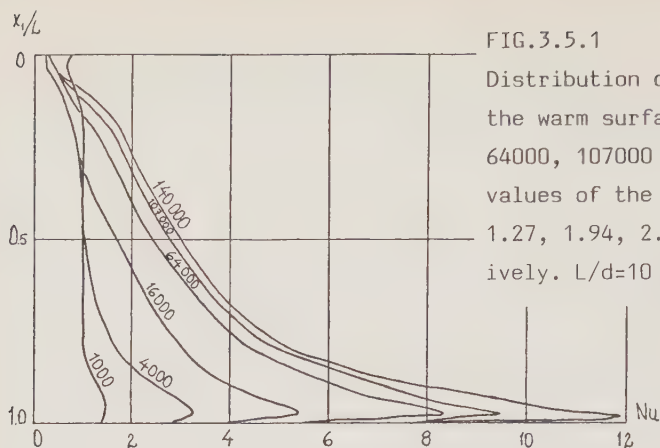


FIG.3.5.1

Distribution of the Nusselt number along the warm surface for $Ra=1000, 4000, 16000, 64000, 107000$ and 140000 . The average values of the Nusselt number are 1.02, 1.27, 1.94, 2.94, 3.39 and 3.68 respectively. $L/d=10$ and $Pr=0.715$.

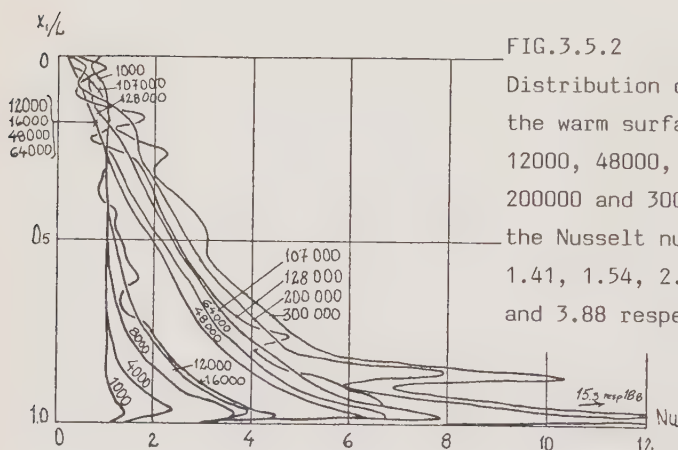


FIG.3.5.2

Distribution of the Nusselt number along the warm surface for $Ra=1000, 4000, 8000, 12000, 48000, 64000, 107000, 128000, 200000$ and 300000 . The average values of the Nusselt number are 1.01, 1.09, 1.26, 1.41, 1.54, 2.16, 2.34, 2.70, 2.80, 3.43 and 3.88 respectively. $L/d=20$ and $Pr=0.715$.

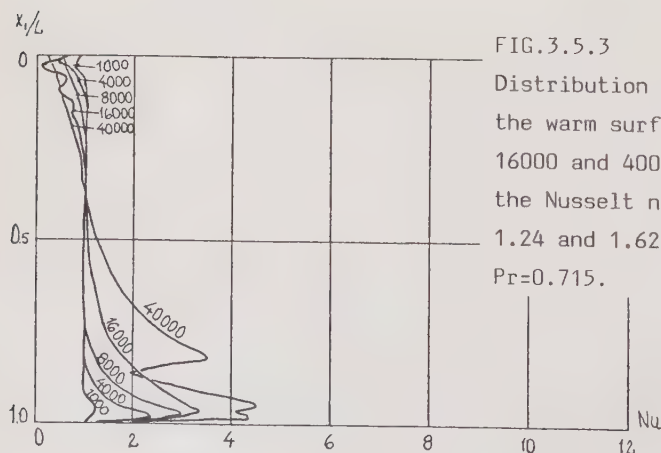


FIG.3.5.3

Distribution of the Nusselt number along the warm surface for $Ra=1000, 4000, 8000, 16000$ and 40000 . The average values of the Nusselt number are 1.00, 1.04, 1.12, 1.24 and 1.62 respectively. $L/d=40$ and $Pr=0.715$.

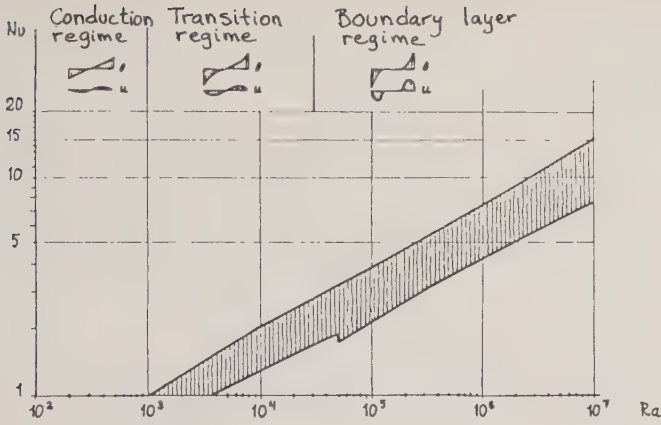


FIG.3.5.4 The Nusselt number as a function of the Rayleigh number for $5 < L/d < 40$, according to Anderlind (1972). A summary of the values reported in the literature (TAB.3.5.1).

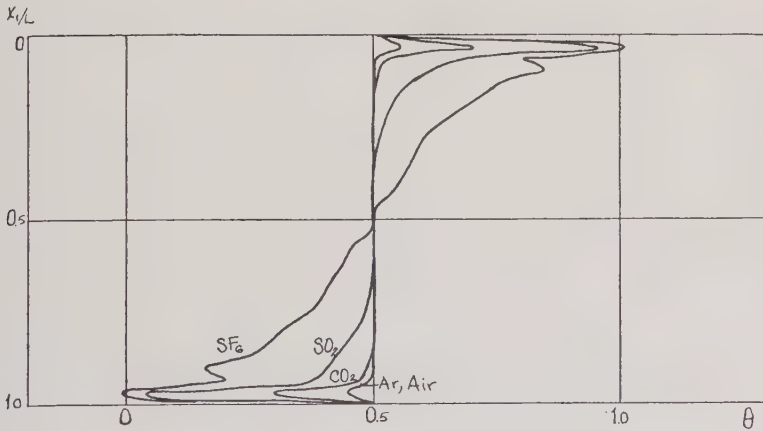


FIG.3.6.1 Vertical temperature distribution for $y_1/d=0.50$, $L/d=75$ and other data according to TAB.3.6.1. $Ra=1250$ (air), 1530 (CO_2), 5140 (CO_2), 17560 (SO_2) and 44280 (SF_6).

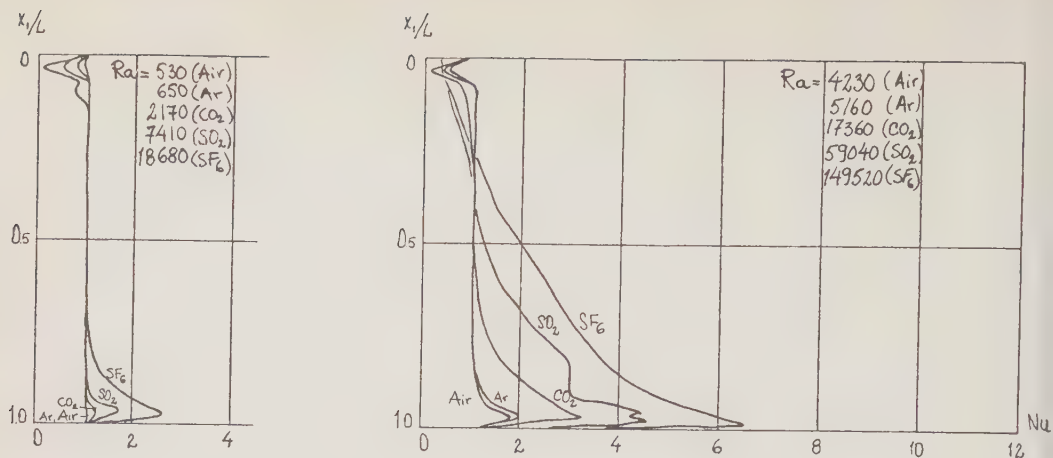


FIG.3.6.2-3.6.3 Distribution of the Nusselt number along the vertical centre line on the warm surface. $L/d=100$ (left) and $L/d=50$ (right) for air, Ar, CO_2 , SO_2 and SF_6 . Other data according to TAB.3.6.1.

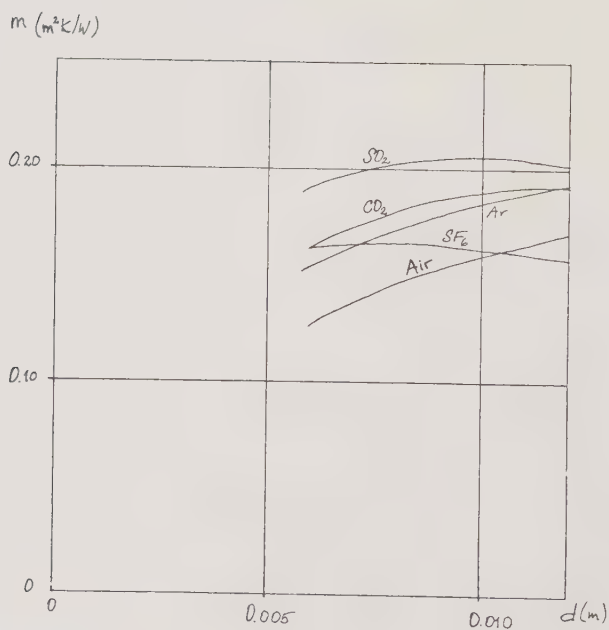


FIG.3.6.4 Calculated thermal resistance with values as in TAB.3.6.2 and $\alpha_R=3.8 \text{ W/m}^2\text{K}$.

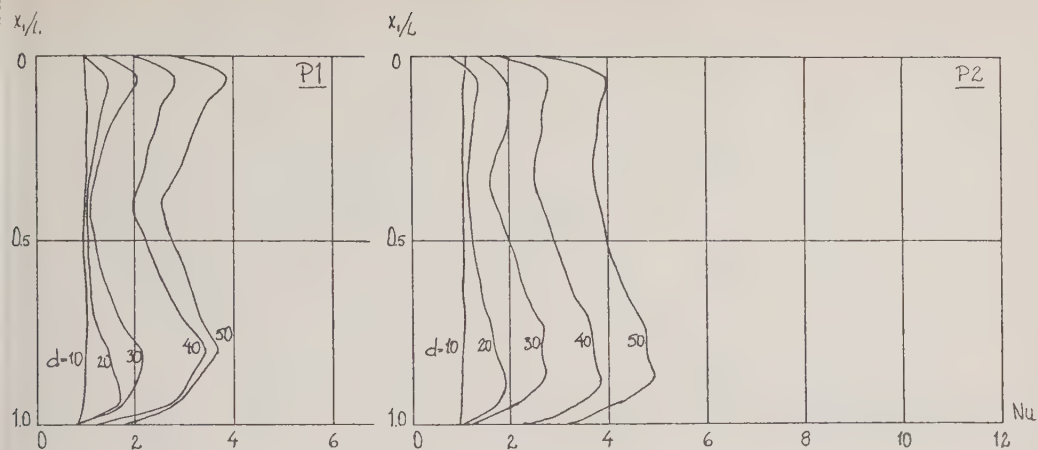


FIG.3.7.1-3.7.2 Distribution of the local Nusselt number along the warm vertical edge in test window P1 and P2 with boundary conditions according to APP.6.2.6-6.2.7. The average values of the Nusselt number were 1.01, 1.26, 1.57, 2.60 and 3.14 for P1 and 1.03, 1.37, 2.21, 2.96 and 4.05 for P2.

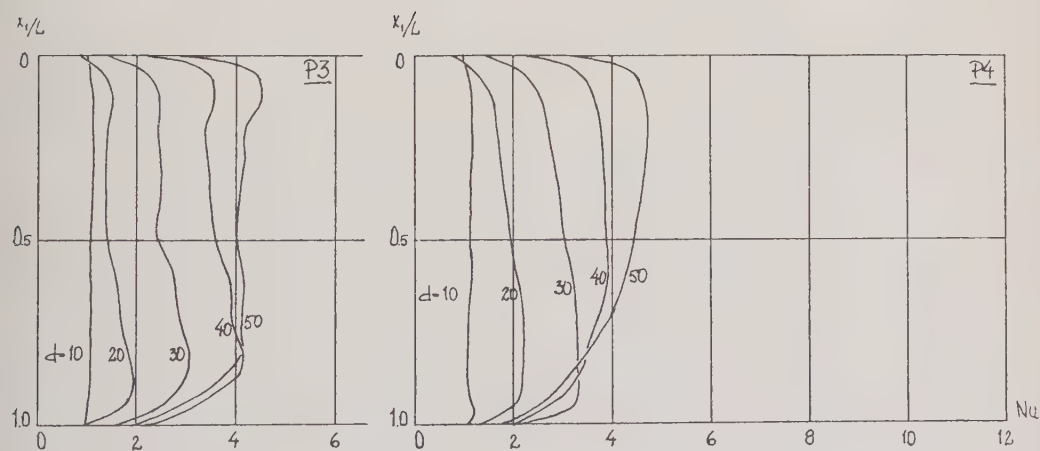


FIG.3.7.3-3.7.4 Distribution of the local Nusselt number along the warm vertical edge in test window P3 and P4 with boundary conditions according to APP.6.2.8-6.2.9. The average values of the Nusselt number were 1.06, 1.53, 2.64, 3.59 and 3.96 for P3 and 1.09, 1.85, 2.91, 3.56 and 4.03 for P4.

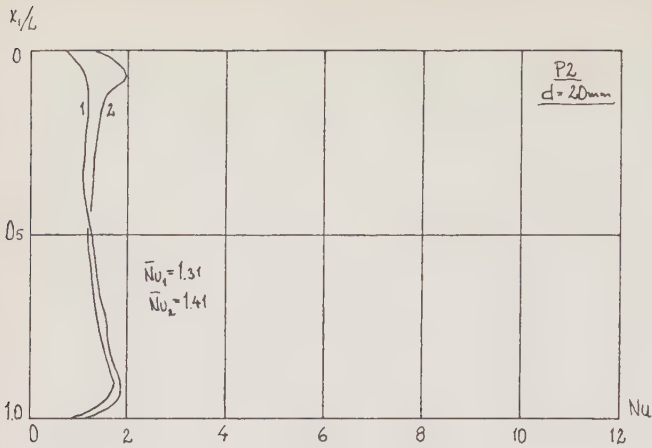


FIG.3.7.5

Vertical distribution of the local Nusselt number along the inside (1) and along the intermediate sheet (2) in test window P2 with boundary conditions according to APP.6.2.10.

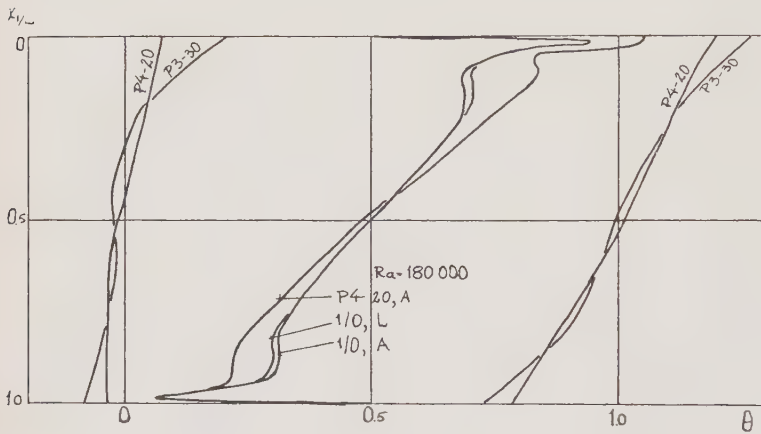


FIG.3.7.6 Vertical temperature distribution for $Ra=180000$ and $L/d=10$. The temperature along the vertical edges is 1/0. A=adiabatic horizontal edges, L=linear temperature distribution.

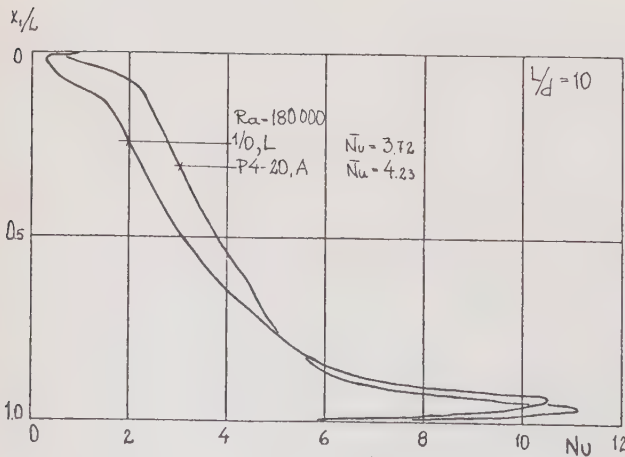


FIG.3.7.7

Distribution of the local Nusselt number along the warm vertical edge for $Ra=180000$ and $L/d=10$. The temperature along the vertical edges is 1/0 or according to PX-XX, see APP. 6.2.6-10. A=adiabatic horizontal edges or L=linear temperature distribution.

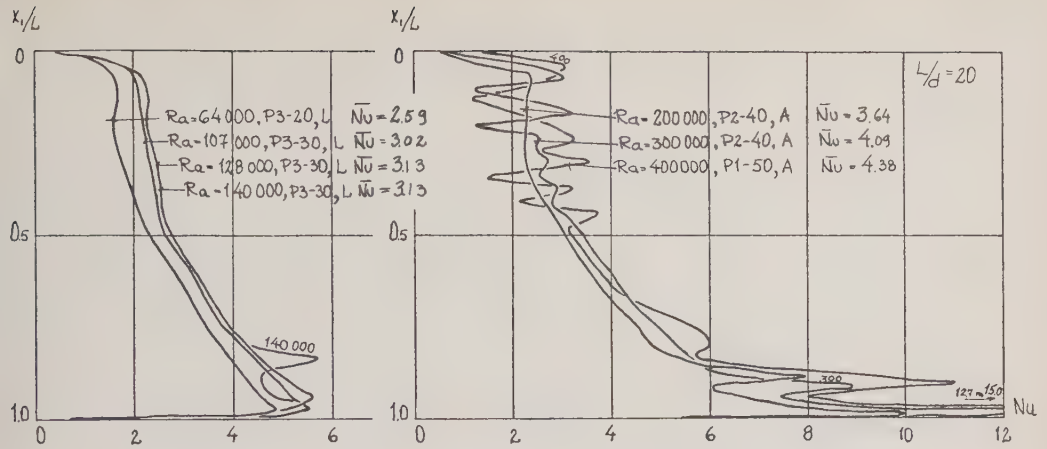


FIG.3.7.8-3.7.9 Distribution of the local Nusselt number along the warm vertical edge for $Ra=64000, 107000, 128000, 140000, 200000, 300000$ and 400000 with $L/d=20$. Temperature along the vertical edges according to PX-XX, see APP.6.2.6-10. A=adiabatic or L=linear temperature distribution along the horizontal edges.

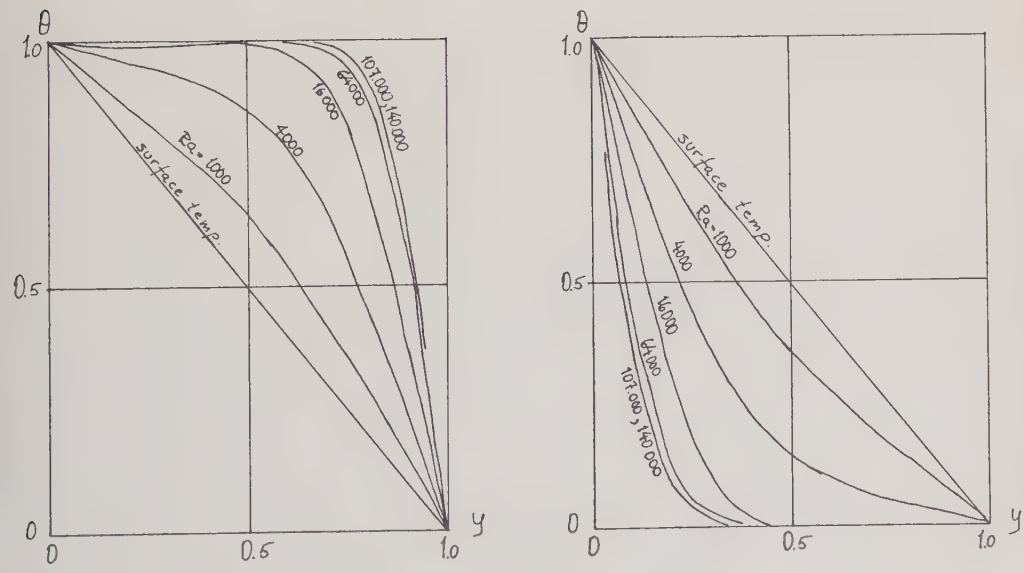


FIG.3.7.10 Surface temperatures and air temperatures at the top and bottom horizontal edge. $L/d=10, 31 \times 17$ mesh, linear temperature distribution along the horizontal edges.

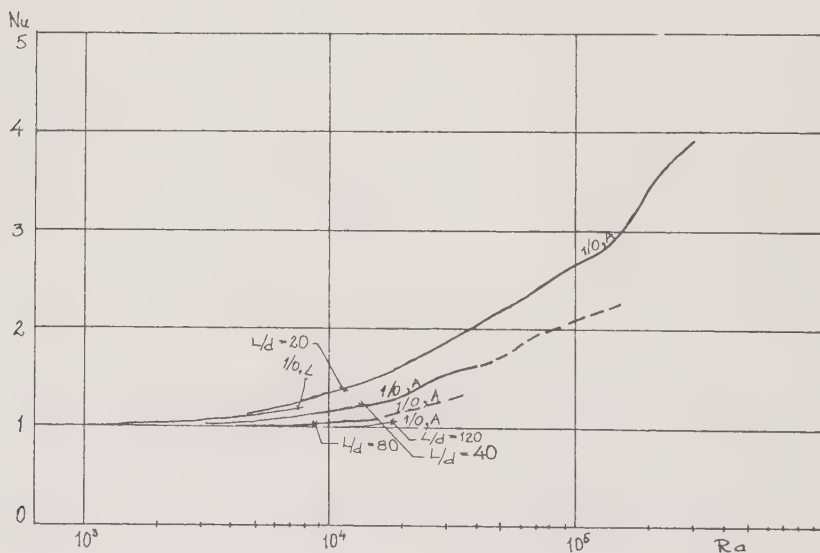
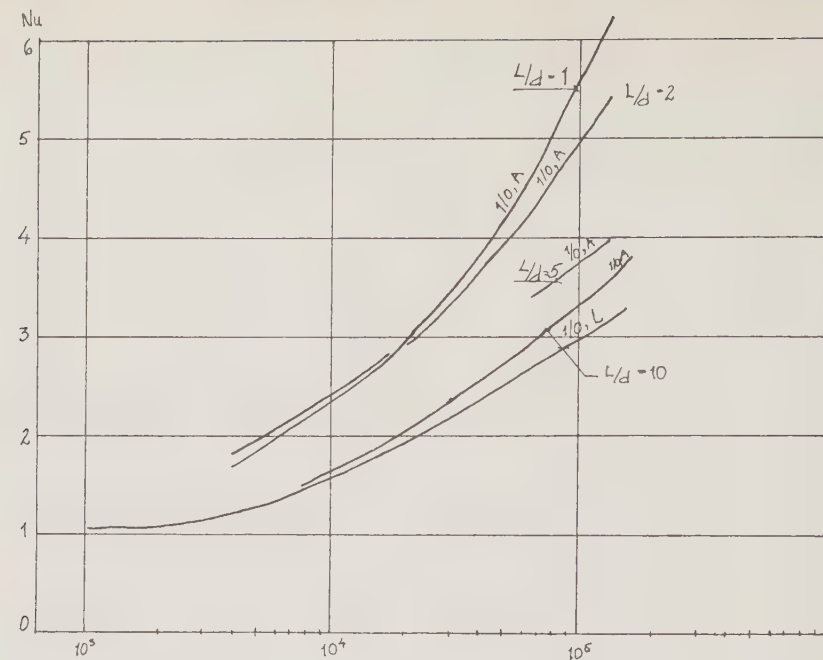


FIG.3.7.11-12 The average Nusselt number as a function of the Rayleigh number for height/width ratios $L/d=1, 2, 5, 10, 20, 40, 80$ and 120 . The temperature along the vertical edges is $1/0$. A=adiabatic horizontal edges or L=linear temperature distribution.

4 NUMERICAL CALCULATIONS OF EDGE FLOWS IN WINDOWS

4.1 GENERAL

When heat transfer is calculated through the edge zones of the glazed portion and adjacent parts such as casement and frame and, in some cases, the wall, the heat transfer must be considered to be two or three dimensional. In addition the construction comprises a number of constituent materials, and the geometrical configuration is often complicated.

An approximate value can be computed by reference to tables or manual methods, but errors are easily introduced since it is difficult to choose the input data in the applied calculation method. Apart from in depth knowledge of the geometrical configuration, the constituent materials and boundary conditions, it is also often necessary for the actual construction to be compared with standard constructions.

Sometimes it is not clear how the stated percentages for glass and frame+casement have been calculated. The area of the frame+casement may be taken to be the area exposed to the indoor air or to the outdoor air, or it may refer to the projection of the window area that can be seen from the inside or outside. The glazed area may refer to the transparent portion of the window, or the glazing rebate may also be included.

In this study, in calculating the overall thermal transmittance k for a window included in the wall element, the window has been projected on to a vertical surface parallel to the window. The areas have been calculated on this projected surface.

4.2 CALCULATION METHOD

The Department of Building Science has developed a program for computation of the temperature distribution under steady two dimensional heat transfer conditions. When the temperature distribution is known, the magnitudes and directions of the heat flows can be calculated. This program (BJ2ST) requires a run time

of 0.5-5 minutes on the Univac 1100 system at the Lund Computer Centre, depending on complexity and the materials chosen for the construction. A mesh is created over the construction by drawing lines parallel to the axes in an orthogonal system of coordinates. The distance between the lines is optional, but the number of meshes in the x or y direction must not exceed 65. This means that the side lengths (dx, dy) of a mesh can be selected at will, but once this has been done all the meshes parallel to the one determined will have one of their sides fixed. The mesh can be refined where large heat fluxes are concentrated or where changes are large. Each mesh can be given a value of the thermal conductivity (λ), and the resistance between adjacent meshes can be specified. The boundary conditions may be fixed temperatures or fixed heat flows. In general, the hole in the wall has been completely insulated, i.e. the heat flow through the frame-wall boundary surface has been zero with the exception of cases 7-3 - 7-7 in which the influence due to an adjacent wall has been studied.

In the case of the upper horizontal surface through the glass and air space, the assumption made has again been that heat transfer through this boundary surface is zero, see e.g. TAB.4.3.1. It has been found in the course of calculations that the isotherms in the upper part have been quite parallel and that the temperature in the upper part of the glazed portion below the boundary surface has been practically constant. Flow at the upper boundary surface has thus been unidimensional, and the thermal transmittance for the midportion of the glazed portion can thus easily be calculated theoretically.

Calculations are carried out by expressing the thermal conductivity equation by finite differences, and solving it by successive iterations according to the forward difference method. When the temperature in all meshes changes by less than 0.01 °C between two successive iterations, the calculation is considered to have converged. The number of iterations has varied between 100 and 6000. The presence of a material of high thermal conductivity (λ) in the construction drastically increases the required number of iterations.

TAB.4.2.2 shows the values of λ used for all the materials em-

ployed. In the case of air, thermal conductivity has been calculated with reference to the resistances quoted in the Swedish Building Code SBN 1980 for non-ventilated vertical air spaces, as set out in TAB.4.2.1. Intermediate values have been obtained by interpolation. All air spaces have thus been assumed to be a homogeneous material with an equivalent value of λ determined with regard to convection and radiation. For an air space of 12 mm, the value of λ has been taken to be 0.090 W/mK. The value $\lambda=0.090$ W/mK has generally been used for the space inside the casement or frame. However, the effect due to variable λ for spaces inside aluminium windows has been studied (see TAB.4.3.19). In order to establish a common reference calculation, a basic case has been calculated for all constructions using the same thermal resistances, namely $m_U=0.055$ m²K/W ($\alpha_U=18$ W/m²K), $m_i=0.115$ m²K/W ($\alpha_i=8.7$ W/m²K), and $m_i+m_U=0.17$ m²K/W. The reason for the choice of this value is that theoretical calculations yield results of this order, and also that measurements in the hot box of the Department show that m_i+m_U for double and triple glazed windows varies between 0.16 and 0.18. Even though the Swedish Building Code stipulates the value $m_i+m_U=0.20$ m²K/W, the values used in the German speaking countries are $\alpha_i=8$ W/m²K and $\alpha_U=23$ W/m²K (for a mean wind velocity of 2 m/s), which gives the value $m_i+m_U=0.17$ m²K/W (DIN 4108).

The above values of $\alpha_i=8.7$ and $\alpha_U=18$ W/m²K may be regarded as representative for the frame or for the stiles and top rail of the casement. For the horizontal bottom rail the values of α differ both for the radiant and convective heat transfer. Special calculations must therefore be made for the bottom rail in order that its temperature distribution may be correctly determined.

The following procedure has been used for calculation of the surface coefficient of heat transfer α_i for the horizontal bottom rail. For the basic case with $m_i=0.115$ m²K/W, the temperatures along the edge are known. In calculating the share due to convection, the expressions

$$\bar{\alpha}_c = 2.2(T_i - T_{is})^{1/4} \quad \text{for a vertical surface}$$

$$\bar{\alpha}_c = 0.6(T_i - T_{is})^{1/4} \quad \text{for downward flow}$$

according to Kollmar, Liese (1957) have been used, where

$$\begin{aligned} T_i &= \text{air temperature (K)} \\ T_{is} &= \text{surface temperature (K)} \end{aligned}$$

With regard to the share due to radiation, it has been assumed that all internal walls are at a temperature of $+20^\circ\text{C}$, and that for the horizontal surface of the bottom rail the window glass occupies one half of the solid angle. We thus have,

for the vertical surface

$$\bar{\alpha}_R = \sigma \epsilon_{\text{RES}} (T_1^4 - T_2^4) / (T_1 - T_2)$$

for the surface next to the glass

$$\bar{\alpha}_R = \sigma \epsilon_{\text{RES}} (T_1^4 - 0.5T_2^4 - 0.5T_3^4) / (T_1 - T_2)$$

where

$$\begin{aligned} T_1 &= \text{temperature of the surface (K)} \\ T_2 &= \text{ambient temperature (293 K)} \\ T_3 &= \text{surface temperature of glass (K)} \\ \sigma &= \text{Stefan-Boltzman constant } (5.67 \cdot 10^{-8} \text{ W/m}^2\text{K}^4) \\ \epsilon_{\text{RES}} &= \text{resultant emissivity} \end{aligned}$$

If surface A_1 is surrounded by surface A_2 , we have

$$\frac{1}{\epsilon_{\text{RES}}} = \frac{1}{\epsilon_1} + \frac{A_1}{A_2} \left(\frac{1}{\epsilon_2} - 1 \right)$$

For $A_1 \ll A_2$, this means that $\epsilon_{\text{RES}} \approx \epsilon_1$.

If the temperatures are known, the values of $\bar{\alpha}_C$ and $\bar{\alpha}_R$ and thus of $m_i = 1/(\bar{\alpha}_C + \bar{\alpha}_R)$ can be calculated. This thermal resistance is used for a new calculation of the temperature distribution. This in turn makes possible renewed calculation of $m_i = 1/(\bar{\alpha}_C + \bar{\alpha}_R)$. This procedure is repeated until the input data and the calculated value of m_i agree. This procedure has

been applied for all basic cases, and the cases described (e.g. 1-8, 3-3, 6-2 etc) thus have the same value for the input data and the calculated m_i . In calculations for a vertical surface with surface temperatures of 278-288 K, we get $\bar{\alpha}_C + \bar{\alpha}_R = 9.1-8.3$ W/m²K and $m_i = 0.11-0.12$ m²K/W, i.e. there is good agreement with the value used for a vertical surface ($m_i = 0.115$ m²K/W).

4.3 EFFECTS DUE TO MATERIALS, BOUNDARY CONDITIONS, CONSTRUCTION

The results of calculations for fixed and openable windows, different material combinations, different glass assembly designs, variable boundary conditions etc are set out in TAB.4.3.1-4.3.20. TAB.4.2.2 sets out the forms of cross hatching used to represent the different materials in TAB.4.3.1-4.3.20. The thermal transmittance through the glazed portion has been calculated according to

$$\frac{1}{k_{gl}} = m_i + m_u + \sum_{gl} \frac{d}{\lambda} + \sum_{space} \frac{d}{\lambda}$$

where

- k_{gl} = thermal transmittance of glazed portion (W/m²K)
- $m_i + m_u$ = internal and external surface resistance (m²K/W)
- d, λ = thickness (m) and thermal conductivity (W/mK)
respectively of the different material layers

Values according to TAB.4.2.1 are used for the layer of air. The heat flow and the temperature in TAB.4.3.1-4.3.20 have been calculated for the appropriate inner portion which is visible from the room. The specific flow for an individual part of the window has been defined as the heat flow (W/m²) through the part concerned divided by the difference in temperature between the inside air and the area of the part. The areas of the glass and the part of the frame concerned have been defined as the internal window area projected on to a vertical plane. The thermal transmittance has been calculated as the sum of the power (W) passing through the inner surfaces divided by the temperature difference and the total projected area. In this way the mean

value (k_{AD}/k_{AF}) is situated between the minimum and maximum values for the part surfaces. Calculation of the thermal transmittance of the window is based on the values according to TAB.4.3.1-4.3.20 and the value of the glass surface weighted with regard to the respective proportions of the surface

$$k_w = k_{gl} \cdot A_{gl}/A_w + k_A \cdot A_A/A_w$$

where

k_w, A_w = thermal transmittance (W/m^2K) and area (m^2) of the window

k_A, A_A = thermal transmittance (W/m^2K) and area (m^2) according to $k_{AD}(k_{AF})$ and $A_{AD}(A_{AF})$ respectively in TAB.4.3.1-4.3.20

k_{gl}, A_{gl} = thermal transmittance (W/m^2K) and area (m^2) of the glass

Note that a 100 mm strip around the edge of the glass forms part of the area $A_{AD}(A_{AF})$.

For instance, for a window of the overall dimensions 1.0x1.0 m and a construction according to TAB.4.3.16, the values will be as follows:

$$A_w = 1.0 \cdot 1.0 = 1.0 \text{ m}^2$$

$$A_{gl} = (1.0 - 2 \cdot 0.1 - 2 \cdot 0.112)^2 = 0.332 \text{ m}^2$$

$$A_A = 1.0 \cdot 1.0 - 0.332 = 0.668 \text{ m}^2$$

$$k_w = 2.21 \cdot 0.332 / 1.00 + 2.65 \cdot 0.668 / 1.00 = 2.50 \text{ W/m}^2K$$

The thermal transmittance for the casement and frame of the construction can be calculated in the same way. If the values selected for the inner or outer thermal resistances are not considered appropriate for a certain case, the thermal transmittance can easily be modified according to

$$\frac{1}{k_{new}} = \frac{1}{k_{tab}} + m^{new} - m^{tab}$$

where

$k_{\text{new}}; m^{\text{new}}$ = new calculated thermal transmittance
with inner thermal resistance m^{new}
 $k_{\text{tab}}; m^{\text{tab}}$ = thermal transmittance and thermal
resistance according to TAB.4.3.1-4.3.20

It is evident from TAB.4.3.1-4.3.20 that it is mainly the construction at the bottom rail of the casement and frame that has been studied. Differences in relation to the stiles or top rail may be the design of the frame, the use of a sealed unit, or the use of aluminium sections in the frame (see cases 6-1, 6-2, 6-3, 6-4, 18-5, 22-1, 22-2). In addition, the internal surface resistance for surfaces B-C according to the tables is different for the bottom rail, stile and top rail. In order therefore that the window model may be as correct as possible, a very large number of calculation cases is required which is both labour and time consuming. In this case it has been decided to study a general case and to carry out calculations for that. For calculation of the thermal transmittance k_w for the 1x1 m window, the design according to the appropriate case has been used around the entire window. Obviously, this is not quite correct, but it facilitates calculations.

FIG.4.3.1-4.3.6 show the temperature profile along the inner surface of the window (not projected), and in FIG.4.3.7-4.3.9 isotherms have been drawn for some interesting cases.

In TAB.4.3.1 and FIG.4.3.1 for cases 1-1 to 1-6, the material in the homogeneous spacer strip has been varied from $\lambda=0.040$ W/mK (thermal insulation material) to $\lambda=210$ W/mK (aluminium). The thermal transmittance k_{AD} increases from 2.75 to 3.01 W/m²K the mean temperature of BC changes by about 2 °C, but the surface temperature in the glassframe corner drops from 5.7 °C to 0.3 °C. If the internal surface resistance of BC is changed from 0.115 (case 1-1) to 0.42 m²K/W (case 1-8) the thermal transmittance k_{AD} decreases by only 0.07 W/m²K, i.e. approx 3%. On the other hand, the mean temperature of BC changes from 13.4 to 9.9 °C. If the pane is placed further out towards the outside air, this produces only marginal changes. FIG.4.3.7 gives

isotherms for case 1-1.

The influence due to different hollow profiles for a glued sealed unit can be seen from TAB.4.3.2 for cases 3-1 - 3-5. On comparing the case with a hollow profile (case 3-1) and a homogeneous aluminium profile (case 1-6), it is seen that these cases are practically identical. The same results are obtained for other materials (lead, plastics). Whether the material in the spacer strip is lead or aluminium makes little difference, but the change is somewhat greater if the material in the spacer strip is PVC.

TAB.4.3.3 for cases 6-1 - 6-5 shows the influence due to an external glazing bead. Constructions with a glazing bead (case 6-5) have thermal transmittance k_w inferior to that for constructions without a glazing bead (case 1-1). It is evident from case 6-1 that the aluminium window flashing below the sealed unit considerably increases the thermal transmittance. In FIG.4.3.8 isotherms for case 6-1 have been plotted.

In TAB.4.3.4 calculations have been carried out for a fixed window. It is evident from these cases that a well insulated wall (e.g. a stud wall with mineral wool) with $\lambda=0.050$ W/mK (case 7-3) is equivalent to insulation around the window construction (case 7-1). On the other hand, lightweight concrete walls with $\lambda=0.12$ W/mK (case 7-4) exhibit very little influence, and a brick wall with $\lambda=0.60$ W/mK (case 7-5) a little larger influence, on the thermal transmittance of the frame (in the latter case about 30%). Isotherms for case 7-1 are plotted in FIG.4.3.9.

TAB.4.3.5-4.3.7 and FIG.4.3.2 for cases 9, 11 and 12 show different metal constructions. The placing of the sealed unit (comparison of cases 9-1 and 12-1) and the choice of material (aluminium/steel) (comparison of cases 9-1 and 9-5) has insignificant influence on the thermal transmittance, in the latter case about 1% of the thermal transmittance k_w . When the sealed unit is placed on the outside (case 9-1), the temperature at corner B (ϑ_B) is higher than when the sealed unit has a more central placing (case 12-1), i.e. 7.1 and 5.8 °C respectively. Case 9-4 represents the internal surface coefficient of heat transfer for a bright surface with an emissivity of approx. 0.1. This naturally reduces

the thermal transmittance, but the surface temperatures on the frame also drop to a very low level. The need for a thermally broken construction is illustrated by a comparison of cases 11-1 and 12-1, there being a change in the thermal transmittance k_w from 4.22 to 3.33 W/m²K.

TAB.4.3.8 gives data for different plastics windows. If reinforcement with steel profiles is necessary, the thermal transmittance k_w is raised by about 4-5%.

In TAB.4.3.9 and FIG.4.3.3 the influence due to different distances between panes and to different glazing rebate depths (d_2) have been studied. Alteration of the width of air space (d_1) has very little effect on temperature distribution in the frame. A change in the depth of glazing rebate (d_2) has most effect on the temperature in the glass-frame corner; when d_2 is increased from 10 mm to 16 mm, this rises from 0.2 (case 16-5) to 2.6 °C (case 16-7). In TAB.4.3.10 the previous construction has been fitted with a glazing bead on the outside. This increases the thermal transmittance a little (comparison of cases 16-2 and 17-1 or 16-9 and 17-2), and the temperature in corner B drops by about 3 °C.

TAB.4.3.11-4.3.15 and FIG.4.3.4-4.3.5 show different constructions for an inward opening window. A comparison of double glazed windows, one with a coupled wooden casement (case 19) and the other with an aluminium casement on the outside (case 20), shows that the thermal transmittance is higher in the latter case (about 3% for the whole window), the main reason being increased flow of heat through the casement. For case 18, 21 and 22 with different triple glazed constructions, the thermal transmittances k_{AF} are 1.96, 2.27 and 2.71 W/m²K respectively, and for the window they are 1.97, 2.22 and 2.54 W/m²K respectively. Note that some of this difference is due to differences in the resistance across the glazed portion. If this is taken into consideration, the difference between cases 18 and 22 is reduced from 0.75 to about 0.65 W/m²K for the part AF. An aluminium strip between frame and casement increases the thermal transmittance k_{AF} by about 0.15 W/m²K and for the window as a whole by about 5% (comparison of cases 18-1 and 18-5 and 22-1 and 22-3).

Different outward opening window constructions are shown in TAB.4.3.16-4.3.18 and FIG.4.3.6. The difference in temperature and thermal transmittance between cases 23 and 24 (with aluminium cladding) is negligible. On the other hand, case 25 has a much more uniform temperature profile and a lower thermal transmittance than case 23 (thermal transmittance $k_w=2.13$ and 2.50 respectively). A comparison of the inward opening and outward opening window (case 21 and 25, and 22 and 23) shows that the thermal transmittance for the outward opening window is a little lower.

TAB.4.3.19-4.3.20 refer to two different types of aluminium window. The thermal transmittances are about the same, but the surface temperature in case 27 is $3-4$ °C lower. The way the assumed value of λ for the space affects the construction is illustrated in cases 26-1 - 26-4. The thermal transmittance k_{AF} changes from 2.85 (case 26-4) to 3.08 (case 26-2) W/m^2K , and the surface temperature of the frame increases by about 1.5 °C.

For a space subject only to conduction the value of λ for air is 0.0241 W/mK . If convection is added, the value of λ rises depending on the direction of the heat flux and temperature conditions. This can be expressed in terms of the Nusselt number.

The radiant heat transfer between two parallel surfaces can be calculated from

$$P = \epsilon_{RES} \sigma A (T_H^4 - T_C^4)$$

Assuming $T_m=273$ K and applying a correction for the side surfaces which are assumed to have the temperature $(T_H+T_C)/2$, we have $\alpha_R=3.3 \epsilon_{RES}$.

The equivalent value of λ is then

$$\lambda_{EKV} = 3.3 \epsilon_{RES}^{d+Nu} \lambda$$

where

d = width of air space (m)

λ = thermal conductivity of air (W/mK)

ϵ_{RES} = resultant emissivity

Nu = Nusselt number

The lowest value for an air space is obtained if the surfaces are bright metal of low emissivity, $\epsilon_{RES}=0.05-0.1$. Taking $d=0.01-0.04$ m and $Nu=1.0$, $\lambda_{EKV}=0.025-0.035$ W/mK. A relatively wide space with matt surfaces ($\epsilon_{RES}=0.7$) will have a higher value of λ_{EKV} . Taking $d=0.03-0.05$ m and $Nu=1.0$, $\lambda_{EKV}=0.11-0.14$ W/mK. If convection increases ($Nu>1$), the value of λ_{EKV} further increases.

A constant value of λ has been assumed for the space between the panes of glass. This represents the case when there is little convection, i.e. the conduction regime. When the temperature difference or the width of space increases, convection increases. Owing to the circulatory air movement in the space, cold air is transferred to the warm side at the bottom corner. There is therefore a risk that convection will cause the surface temperature to drop further at the critical corner between glass and casement. A comparison with a wooden spacer, for case 1-3 (1-8) in TAB.4.3.1 and case P6+L10+G4 in APP.6.2.2, gives the temperatures 4.5°C (2.6°C) and about 4.0°C respectively. For a (hollow) aluminium profile, a comparison is made between case 3-1 (3-3) in TAB.4.3.2 and case D4-12 in APP.6.4.1. The temperatures are 0.3°C (-1.2°C) and -0.8°C respectively. For these cases the Rayleigh number is approximately 10000, and the temperatures for the calculated and measured cases are in good agreement.

A comparison of cases with a constant internal surface coefficient of heat transfer ($\alpha_i=8.7 \text{ W/m}^2\text{K}$) and cases with a calculated value of α_i for the horizontal portion of the bottom rail (part BC) shows that the temperature at point B (ϑ_B) is reduced by $1.5-2.2^{\circ}\text{C}$. During measurements (Sections 6.2, 6.3) on test windows P1-P6, the temperature (ϑ_{is}) in the lower part of the air space was locally reduced by $2.0-2.5^{\circ}\text{C}$. This shows that

the high value of α_i for the horizontal part is on its own sufficient to cause a relatively large reduction in minimum temperature.

Measurements (Section 6.4) on test window P7 with sealed glass units show that the difference in temperature between the corner and a point about 100 mm inside the glass surface ($x_1/L=0.88$) is approximately $4.6-6^\circ\text{C}$. A comparison with a calculated case (TAB.4.3.3) shows that the temperature in the corner is about $5.5-6.4^\circ\text{C}$ lower than at a point some distance towards the middle. It is also evident that the gas used exerts relatively little influence on this temperature difference. A comparison between FIG.6.4.4 (SF_6) and FIG.6.4.5 (Argon) shows that there is no difference in spite of the fact that the Rayleigh number is approx 150000 and 6000 respectively. When comparisons are made between calculated and measured cases, it is obviously important that the cases studied should be similar. The way the sealed unit is mounted has great significance for the minimum temperature.

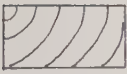
It will be seen from TAB.4.3.1-4.3.20 that the internal surface coefficient of heat transfer for the horizontal part of the frame or casement varies with the type of construction and the materials used. For frame and casement of wood or plastics, $m_i=0.32-0.47$ for a double glazed window, with the mean value $m_i=0.40\text{ m}^2\text{K/W}$, and for triple glazed window $m_i=0.25-0.35$, with the mean value $m_i=0.29\text{ m}^2\text{K/W}$. For an aluminium frame and casement, $m_i=0.30\text{ m}^2\text{K/W}$ for a double glazed window and $m_i=0.26\text{ m}^2\text{K/W}$ for a triple glazed window. When the frame-casement is not insulated, $m_i=0.22\text{ m}^2\text{K/W}$.

In these calculations, the thermal transmittance for a $1\times 1\text{ m}$ window varied between 1.97 and $2.54\text{ W/m}^2\text{K}$ (openable triple glazed window), between 2.76 and $2.84\text{ W/m}^2\text{K}$ (openable double glazed window), between 2.61 and $2.72\text{ W/m}^2\text{K}$ (openable triple glazed aluminium window), and between 2.12 and $2.73\text{ W/m}^2\text{K}$ (openable triple glazed plastics window). For fixed wooden windows with a sealed glass unit, the values are approximately $3.0\text{ W/m}^2\text{K}$ for a double glazed window and $2.2-2.3\text{ W/m}^2\text{K}$ for a triple glazed window.

TAB.4.2.1 Thermal resistance of an unventilated vertical air space (according to the Swedish Building Code SBN 1980).

Air space (mm)	Thermal resistance (m^2K/W)	Thermal conductivity (W/mK)
5	0.10	0.050
10	0.13	0.077
20	0.15	0.133
50	0.16	0.312
100	0.16	0.625

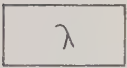
TAB.4.2.2 Thermal conductivity (W/mK) and designations used for materials in TAB.4.3.1-4.3.20.



Wood
 $\lambda=0.14$ W/mK



Glass
 $\lambda=0.8$ W/mK



Air or other material
 λ acc. TAB.4



Aluminium
 $\lambda=210.0$ W/mK



Plastics (PVC)
 $\lambda=0.16$ W/mK



Steel
 $\lambda=60.0$ W/mK



Plastics
 $\lambda=0.20$ W/mK



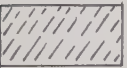
Lead
 $\lambda=35.0$ W/mK



Polyurethane foam
 $\lambda=0.030$ W/mK

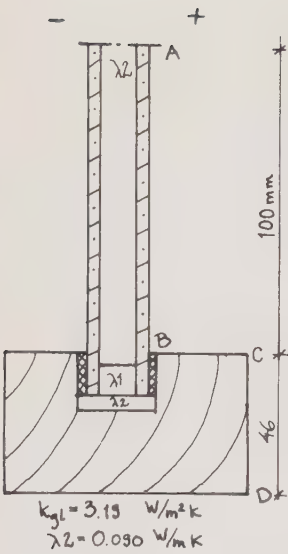


Rubber
 $\lambda=0.12$ W/mK



Polystyrene
 $\lambda=0.040$ W/mK

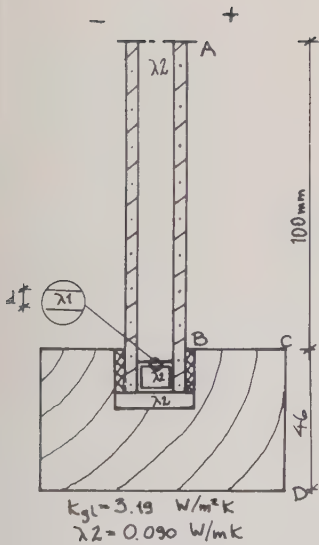
TAB.4.3.1 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of a fixed window. Wooden frame. $\vartheta_U = -20^\circ\text{C}$, $\vartheta_i = 20^\circ\text{C}$, $m_U = 0.055$ and $m_i = 0.115 \text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of frame according to Case 1-8. Assumed surface resistance for stile of frame according to Case 1-7. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
1-1 $\lambda_1 = 0.16$	AB	5.2	128.4	3.21
	BC	13.4	55.2	1.38
	CD	15.4	39.2	0.98
1-2 $\lambda_1 = 0.040$	AB	5.3	126.8	3.17
	BC	14.1	50.0	1.25
	CD	15.6	37.2	0.93
1-3 $\lambda_1 = 0.14$	AB	5.2	128.2	3.20
	BC	13.5	54.5	1.36
	CD	15.4	39.0	0.97
1-4 $\lambda_1 = 0.80$	AB	4.9	130.9	3.27
	BC	12.5	62.9	1.57
	CD	15.0	42.3	1.06
1-5 $\lambda_1 = 35.0$	AB	4.6	132.8	3.32
	BC	11.9	68.3	1.71
	CD	14.7	44.3	1.11
1-6 $\lambda_1 = 210.0$	AB	4.6	132.9	3.32
	BC	11.9	58.5	1.71
	CD	14.7	44.4	1.11
1-7 $\lambda_1 = 0.16$ $m_{BC} = 0.20$	AB	5.0	129.5	3.24
	BC	11.8	40.2	1.01
	CD	14.8	43.6	1.09
1-8 $\lambda_1 = 0.16$ $m_{BC} = 0.42$	AB	4.9	130.6	3.26
	BC	9.9	23.8	0.59
	CD	14.2	48.6	1.21

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^\circ\text{C}$)
	k_{AD}	k_{BD}	k_W	
1-1	2.82	1.97	3.01	4.3
1-2	2.75	1.82	2.97	5.7
1-3	2.81	1.95	3.00	4.5
1-4	2.93	2.18	3.06	2.0
1-5	3.01	2.33	3.10	0.3
1-6	3.01	2.34	3.10	0.3
1-7	2.79	1.81	2.99	3.5
1-8	2.75	1.64	2.97	2.6

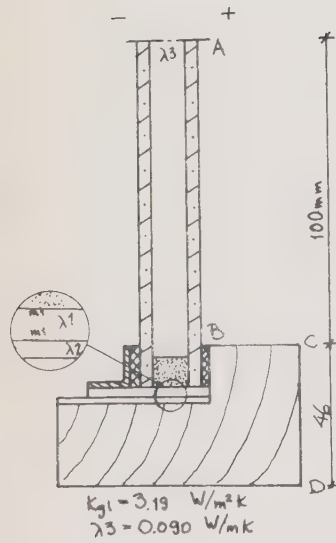
TAB.4.3.2 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of a fixed window. Wooden frame. $\vartheta_u = -20^\circ\text{C}$, $\vartheta_i = 20^\circ\text{C}$, $m_u = 0.055$ and $m_i = 0.115 \text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of frame according to Case 3-3. Assumed surface resistance for stile of frame according to Case 3-2. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
3-1 $\lambda_1 = 210.0$ $d = 1\text{mm}$	AB	4.6	132.8	3.32
	BC	11.9	68.1	1.70
	CD	14.8	44.3	1.11
3-2 $\lambda_1 = 210.0$ $d = 1\text{mm}$ $m_{BC} = 0.20$	AB	4.7	132.6	3.32
	BC	12.0	67.5	1.69
	CD	14.8	44.1	1.10
3-3 $\lambda_1 = 210.0$ $d = 1\text{mm}$ $m_{BC} = 0.36$	AB	4.4	134.9	3.37
	BC	8.2	32.4	0.81
	CD	13.6	53.9	1.35
3-4 $\lambda_1 = 35.0$ $d = 1\text{mm}$	AB	4.5	133.9	3.35
	BC	10.0	49.2	1.23
	CD	14.2	49.3	1.23
3-5 $\lambda_1 = 0.16$ $d = 2\text{mm}$	AB	5.2	128.1	3.20
	BC	13.6	54.0	1.35
	CD	15.4	38.8	0.97

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^\circ\text{C}$)
	k_{AD}	k_{BD}	k_W	
3-1	3.01	2.33	3.10	0.3
3-2	3.00	2.31	3.10	0.5
3-3	2.92	1.93	3.05	-1.2
3-4	2.96	2.11	3.08	-0.5
3-5	2.81	1.94	3.00	4.6

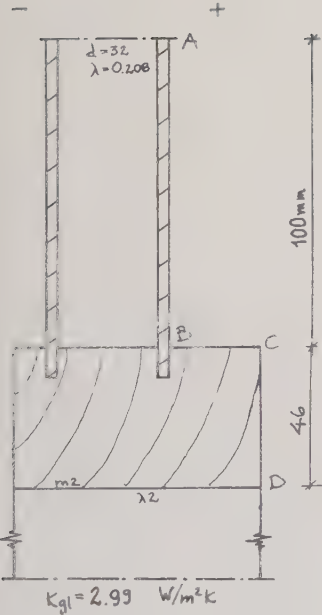
TAB.4.3.3 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of a fixed window. Wooden frame. $\vartheta_U = -20^\circ\text{C}$, $\vartheta_i = 20^\circ\text{C}$, $m_U = 0.055$ and $m_i = 0.115 \text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of frame according to Case 6-2. Material designations according to TAB. 4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
6-1	AB	4.4	134.9	3.37
$\lambda_1 = 0.090$	BC	10.5	79.8	2.00
$\lambda_2 = 210.0$	CD	13.0	59.3	1.48
$m_1 = 0$				
6-2	AB	4.2	136.9	3.42
$\lambda_1 = 0.090$	BC	6.6	41.3	1.03
$\lambda_2 = 210.0$	CD	11.9	68.7	1.72
$m_1 = 0$				
$m_{BC} = 0.32$				
6-3	AB	4.7	132.7	3.32
$\lambda_1 = 0.050$	BC	11.3	73.1	1.83
$\lambda_2 = 210.0$	CD	13.3	56.8	1.42
$m_1 = 0.115$				
6-4	AB	4.4	135.2	3.38
$\lambda_1 = 2.0$	BC	10.4	81.0	2.02
$\lambda_2 = 210.0$	CD	12.7	61.6	1.54
$m_1 = 0.055$				
6-5	AB	5.0	130.0	3.25
$\lambda_1 = 0.090$	BC	13.0	58.7	1.47
$\lambda_2 = 0.14$	CD	15.0	41.9	1.05
$m_1 = 0$				

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^\circ\text{C}$)
	k_{AD}	k_{BD}	k_W	
6-1	3.23	2.91	3.21	-0.1
6-2	3.12	2.46	3.16	-1.7
6-3	3.13	2.73	3.16	1.6
6-4	3.26	2.99	3.22	-0.2
6-5	2.89	2.10	3.04	3.4

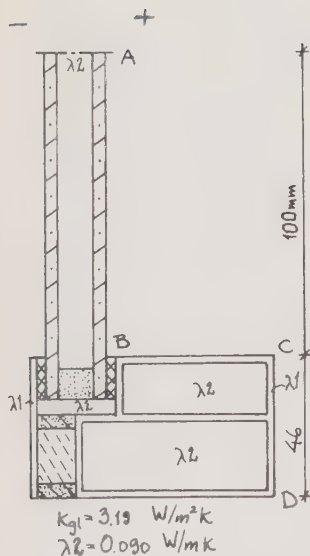
TAB.4.3.4 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of a fixed window. Wooden frame. $\vartheta_u = -20\text{ }^{\circ}\text{C}$, $\vartheta_i = 20\text{ }^{\circ}\text{C}$, $m_u = 0.055$ and $m_i = 0.115\text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of frame according to Case 7-2. Material designations according to TAB. 4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur- face	Mean temp ($^{\circ}\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
7-1 $m_2 = \infty$	AB	6.2	119.3	2.98
	BC	14.2	50.0	1.22
	CD	15.6	36.7	0.92
7-2 $m_2 = \infty$ $m_{BC} = 0.47$	AB	5.9	121.7	3.04
	BC	11.0	18.9	0.47
	CD	14.7	45.1	1.13
7-3 $\lambda_2 = 0.050$ $m_2 = 0$	AB	6.2	119.4	2.99
	BC	14.2	49.0	1.23
	CD	15.8	35.7	0.89
7-4 $\lambda_2 = 0.12$ $m_2 = 0$	AB	6.2	119.6	2.99
	BC	14.1	49.8	1.25
	CD	15.5	38.1	1.16
7-5 $\lambda_2 = 0.60$ $m_2 = 0$	AB	6.1	120.4	3.01
	BC	13.6	54.1	1.35
	CD	13.6	53.5	1.34

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^{\circ}\text{C}$)
	k_{AD}	k_{BD}	k_W	
7-1	2.58	1.72	2.79	6.5
7-2	2.54	2.24	2.77	5.0
7-3	2.58	1.69	2.79	6.5
7-4	2.60	1.77	2.80	6.4
7-5	2.76	2.22	2.88	5.9

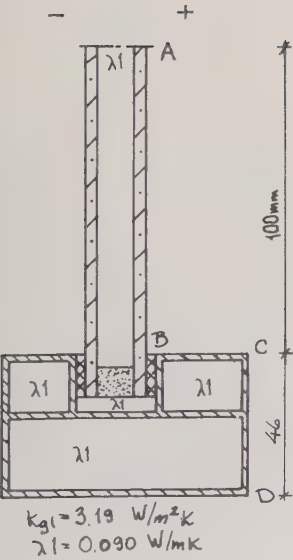
TAB.4.3.5 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of a fixed window. Metal frame. $\vartheta_y = -20^\circ\text{C}$, $\vartheta_i = 20^\circ\text{C}$, $m_U = 0.055$ and $m_i = 0.115 \text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of frame according to Case 9-3. Assumed surface resistance for stile of frame according to Case 9-2. Frame of low emissivity ($\varepsilon \approx 0.1$) according to Case 9-4. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
9-1 $\lambda_1 = 210.0$	AB	5.6	124.4	3.11
	BC	10.8	79.8	2.00
	CD	10.9	79.3	1.98
9-2 $\lambda_1 = 210.0$ $m_{BC} = 0.20$	AB	5.4	126.0	3.15
	BC	9.0	55.1	1.38
	CD	9.1	94.9	2.37
9-3 $\lambda_1 = 210.0$ $m_{BC} = 0.32$	AB	5.3	127.0	3.18
	BC	7.8	38.0	0.95
	CD	8.0	104.7	2.62
9-4 $\lambda_1 = 210.0$ $m_{BC} = 2.0$ $m_{CD} = 0.238$	AB	4.5	134.3	3.36
	BC	1.2	9.4	0.24
	CD	1.2	78.9	1.97
9-5 $\lambda_1 = 60.0$	AB	5.6	124.6	3.12
	BC	11.0	77.8	1.94
	CD	11.2	76.9	1.92

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^\circ\text{C}$)
	k_{AD}	k_{BD}	k_W	
9-1	3.51	4.37	3.35	7.1
9-2	3.43	4.02	3.31	6.0
9-3	3.36	3.75	3.27	5.3
9-4	3.01	2.26	3.10	1.2
9-5	3.47	4.25	3.33	7.0

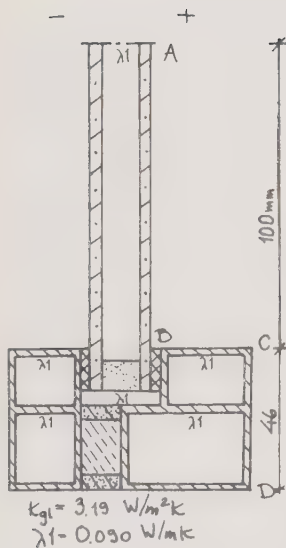
TAB.4.3.6 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of a fixed window. Aluminium frame. $\vartheta_u = -20\text{ }^{\circ}\text{C}$, $\vartheta_i = 20\text{ }^{\circ}\text{C}$, $m_u = 0.055$ and $m_i = 0.115\text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of frame according to Case 11-2. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^{\circ}\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
11-1	AB	4.4	135.2	3.38
	BC	-4.8	215.4	5.39
	CD	-5.2	218.7	5.47
11-2 $m_{BC} = 0.22$	AB	4.1	137.2	3.43
	BC	-6.6	120.7	3.02
	CD	-6.8	233.3	5.83

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^{\circ}\text{C}$)	ϑ_D ($^{\circ}\text{C}$)
	k_{AD}	k_{BD}	k_W		
11-1	5.26	9.33	4.22	-1.3	-5.3
11-2	4.87	8.01	4.03	-2.6	-7.0

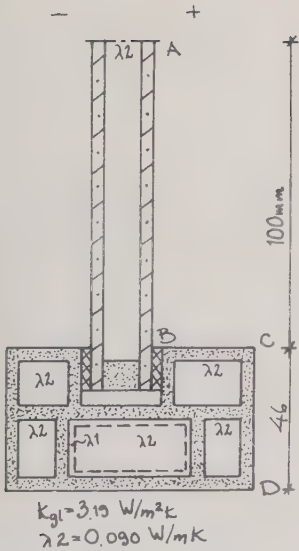
TAB.4.3.7 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of a fixed window. Aluminium frame. $\vartheta_U = -20^\circ\text{C}$, $\vartheta_i = 20^\circ\text{C}$, $m_U = 0.055$ and $m_i = 0.115 \text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of frame according to Case 12-2. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
12-1	AB	5.4	126.4	3.16
	BC	8.8	97.7	2.44
	CD	8.8	97.3	2.43
12-2 $m_{BC} = 0.29$	AB	5.2	128.3	3.21
	BC	6.6	46.2	1.15
	CD	6.7	115.5	2.89

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^\circ\text{C}$)
	k_{AD}	k_{BD}	k_W	
12-1	3.48	4.18	3.33	5.8
12-2	3.37	3.72	3.28	4.5

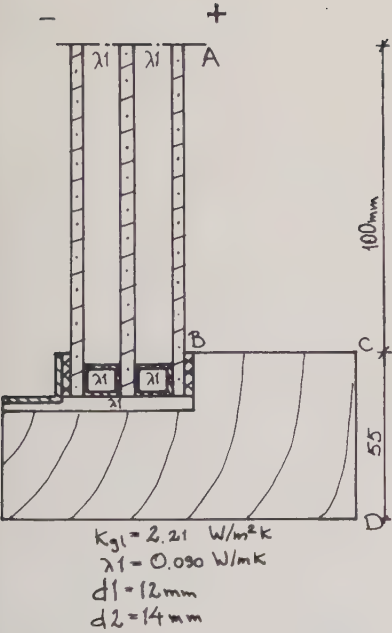
TAB.4.3.8 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of a fixed window. Plastics frame. $\vartheta_U = -20\text{ }^{\circ}\text{C}$, $\vartheta_i = 20\text{ }^{\circ}\text{C}$, $m_U = 0.055$ and $m_i = 0.115\text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of frame according to Case 13-2. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^{\circ}\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
13-1 $\lambda_1 = 0.090$	AB	5.2	128.4	3.21
	BC	13.8	52.9	1.32
	CD	16.0	34.1	0.85
13-2 $\lambda_1 = 0.090$ $m_{BC} = 0.44$	AB	4.9	130.7	3.27
	BC	10.2	22.2	0.56
	CD	14.9	42.8	1.07
13-3 $\lambda_1 = 60.0$	AB	5.1	129.2	3.23
	BC	12.4	64.2	1.61
	CD	13.2	57.5	1.44

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^{\circ}\text{C}$)
	k_{AD}	k_{BD}	k_W	
13-1	2.77	1.80	2.98	4.3
13-2	2.70	1.47	2.95	2.5
13-3	3.03	2.59	3.11	3.5

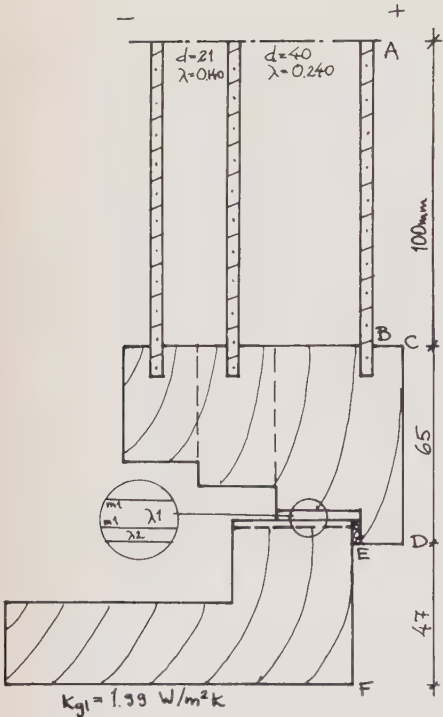
TAB.4.3.10 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of a fixed window. Wooden frame. $\vartheta_{\text{e}}=-20\text{ }^{\circ}\text{C}$, $\vartheta_{\text{i}}=20\text{ }^{\circ}\text{C}$, $m_{\text{U}}=0.055$ and $m_{\text{i}}=0.115\text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of frame according to Case 17-2. Material designations according to TAB. 4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^{\circ}\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
17-1	AB	7.7	106.0	2.65
	BC	13.1	58.5	1.46
	CD	16.6	28.8	0.72
17-2 $m_{\text{BC}}=0.32$	AB	7.5	108.4	2.71
	BC	9.3	33.0	0.82
	CD	15.6	37.1	0.93

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_{B} ($^{\circ}\text{C}$)
	k_{AD}	k_{BD}	k_{W}	
17-1	2.49	2.17	2.36	-1.3
17-2	2.38	1.74	2.30	-2.8

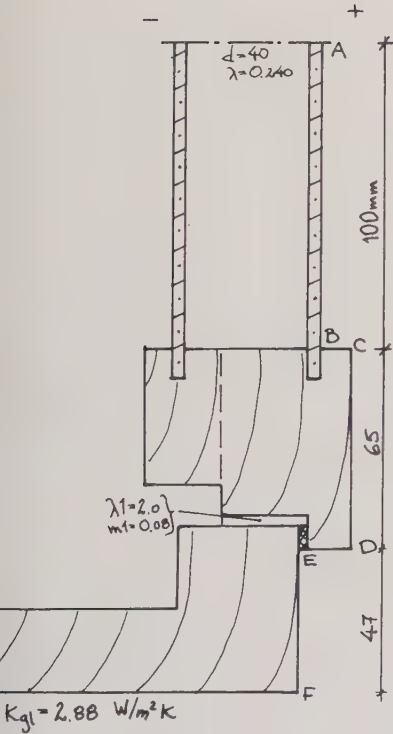
TAB.4.3.11 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of an openable window. Wooden casement and frame. $\vartheta_U = -20\text{ }^{\circ}\text{C}$, $\vartheta_i = 20\text{ }^{\circ}\text{C}$, $m_U = 0.055$ and $m_i = 0.115\text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of sash according to Case 18-4. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^{\circ}\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
18-1	AB	12.8	61.9	1.55
	BC	14.3	48.3	1.21
	CD	12.9	59.8	1.50
	DE	10.5	80.0	2.00
	EF	8.1	100.0	2.50
18-2	AB	13.0	60.1	1.50
	BC	14.7	44.7	1.12
	CD	14.5	46.7	1.17
	DE	13.9	51.7	1.29
	EF	9.6	88.1	2.20
18-3	AB	12.6	63.6	1.59
	BC	13.8	51.9	1.30
	CD	11.2	74.1	1.85
	DE	6.8	111.3	2.78
	EF	6.8	111.7	2.79
18-4	AB	12.7	62.8	1.57
	BC	13.1	22.6	0.57
	CD	12.7	61.5	1.54
	DE	10.5	80.1	2.00
	EF	8.1	100.0	2.50
18-5	AB	12.8	62.3	1.56
	BC	14.2	49.3	1.23
	CD	12.2	65.8	1.65
	DE	7.5	105.3	2.63
	EF	7.1	108.8	2.72

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^{\circ}\text{C}$)	ϑ_E ($^{\circ}\text{C}$)
	k_{AF}	k_{BF}	k_W		
18-1	1.96	2.33	1.97	11.4	4.8
18-2	1.71	1.90	1.80	12.0	8.8
18-3	2.22	2.78	2.14	10.9	-1.0
18-4	1.95	2.30	1.96	10.9	4.0
18-5	2.11	2.61	2.07	11.3	-1.4

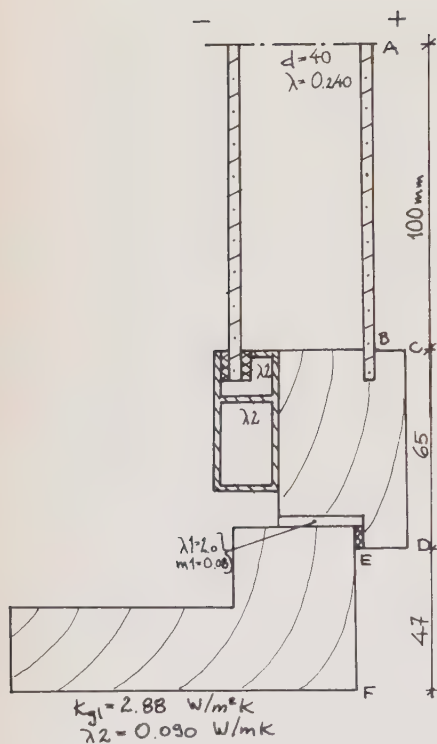
TAB.4.3.12 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of an openable window. Wooden casement and frame. $t_U = -20\text{ }^{\circ}\text{C}$, $t_i = 20\text{ }^{\circ}\text{C}$, $m_U = 0.055$ and $m_i = 0.115\text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of sash according to Case 19-2. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^{\circ}\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
19-1	AB	6.7	115.0	2.88
	BC	11.8	69.4	1.73
	CD	11.7	70.1	1.75
	DE	10.2	82.4	2.06
	EF	8.1	100.5	2.51
19-2	AB	6.6	116.2	2.91
$m_{BC} = 0.36$	BC	9.9	27.7	0.69
	CD	11.4	72.7	1.82
	DE	10.2	82.6	2.07
	EF	8.1	100.5	2.51

Case	k-value ($\text{W/m}^2\text{K}$)			t_B	t_E
	k_{AF}	k_{BF}	k_W	($^{\circ}\text{C}$)	($^{\circ}\text{C}$)
19-1	2.70	2.54	2.76	7.3	3.8
19-2	2.68	2.48	2.75	6.5	3.8

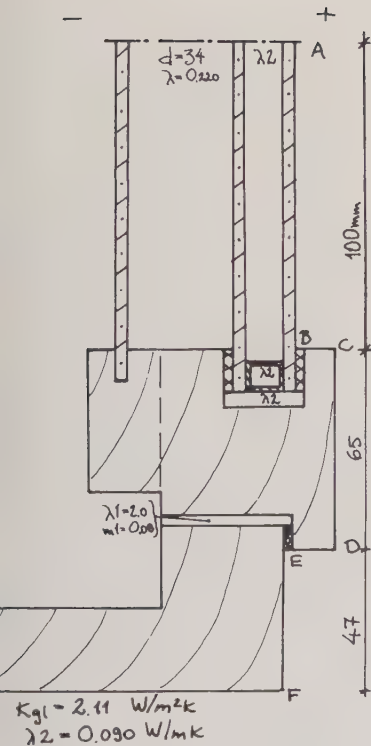
TAB.4 3.13 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of an openable window. Wood+aluminium casement and wooden frame.
 $\vartheta_u = -20^\circ\text{C}$, $\vartheta_i = 20^\circ\text{C}$, $m_u = 0.055$ and $m_i = 0.115$
 $\text{m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of sash according to Case 20-2.
 Material designations according to TAB.4 2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
20-1	AB	6.1	120.4	3.01
	BC	10.8	78.4	1.96
	CD	11.0	76.3	1.91
	DE	10.0	84.2	2.10
	EF	7.4	100.3	2.51
20-2	AB	5.9	121.6	3.04
	BC	8.8	31.9	0.80
	CD	10.6	79.1	1.98
	DE	10.0	84.3	2.11
	EF	7.4	100.3	2.51

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^\circ\text{C}$)
	k_{AF}	k_{BF}	k_W	
20-1	2.82	2.65	2.84	5.7
20-2	2.80	2.59	2.83	4.9

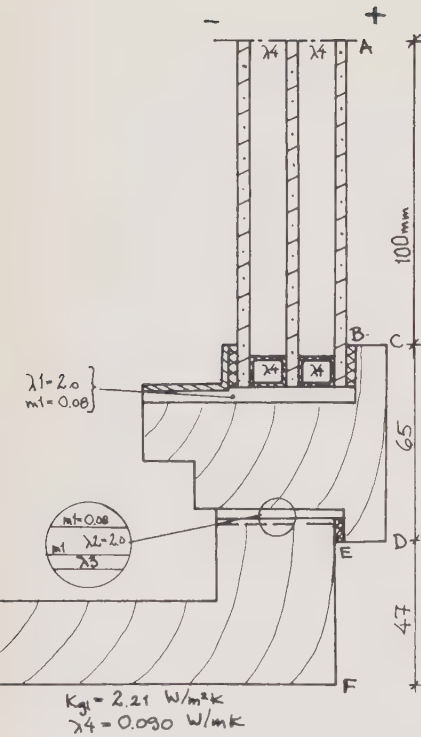
TAB.4.3.14 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of an openable window. Wooden casement and frame. $\vartheta_u = -20\text{ }^{\circ}\text{C}$, $\vartheta_i = 20\text{ }^{\circ}\text{C}$, $m_u = 0.055$ and $m_i = 0.115\text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of sash according to Case 21-2. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^{\circ}\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
21-1	AB	9.9	87.3	2.18
	BC	11.8	68.6	1.72
	CD	12.7	61.9	1.55
	DE	11.6	70.5	1.76
	EF	9.2	93.6	2.34
21-2	AB	9.8	88.6	2.21
	BC	10.0	33.9	0.85
	CD	12.3	64.7	1.62
	DE	11.6	70.8	1.77
	EF	9.2	93.7	2.34

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^{\circ}\text{C}$)
	k_{AF}	k_{BF}	k_W	
21-1	2.27	2.35	2.22	6.7
21-2	2.25	2.29	2.20	6.0

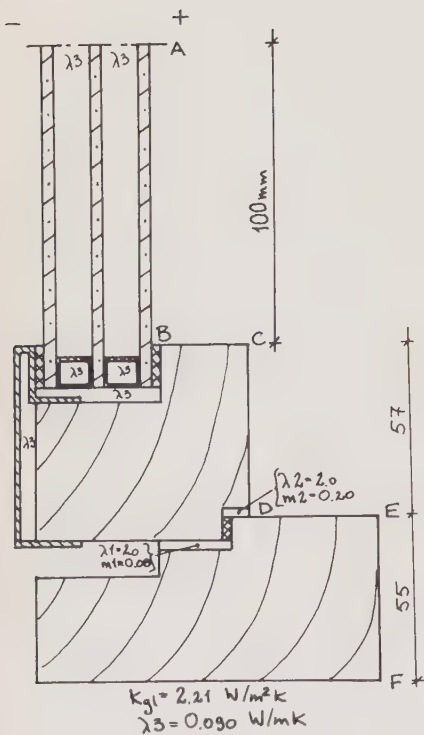
TAB.4.3.15 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of an openable window. Wooden casement and frame. $\dot{q}_U = -20^\circ\text{C}$, $\dot{q}_i = 20^\circ\text{C}$, $m_U = 0.055$ and $m_i = 0.115 \text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of sash according to Case 22-2. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
22-1 $\lambda_3 = 210.0$	AB	7.9	105.1	2.63
	BC	7.1	109.0	2.73
	CD	9.6	88.1	2.20
	DE	8.3	98.6	2.47
	EF	8.1	102.7	2.57
22-2 $m_{BC} = 0.25$ $\lambda_3 = 210.0$	AB	7.7	106.2	2.66
	BC	4.7	60.3	1.51
	CD	9.2	91.2	2.28
	DE	8.3	98.8	2.47
	EF	8.1	102.7	2.57
22-3 $\lambda_3 = 0.14$	AB	7.9	104.9	2.62
	BC	7.1	108.4	2.71
	CD	10.2	82.5	2.06
	DE	11.0	75.5	1.89
	EF	9.0	94.8	2.37

Case	k-value ($\text{W/m}^2\text{K}$)			$\dot{q}_B$ ($^\circ\text{C}$)
	k_{AF}	k_{BF}	k_W	
22-1	2.85	3.05	2.64	-0.8
22-2	2.81	2.95	2.61	-1.6
22-3	2.71	2.79	2.54	-0.7

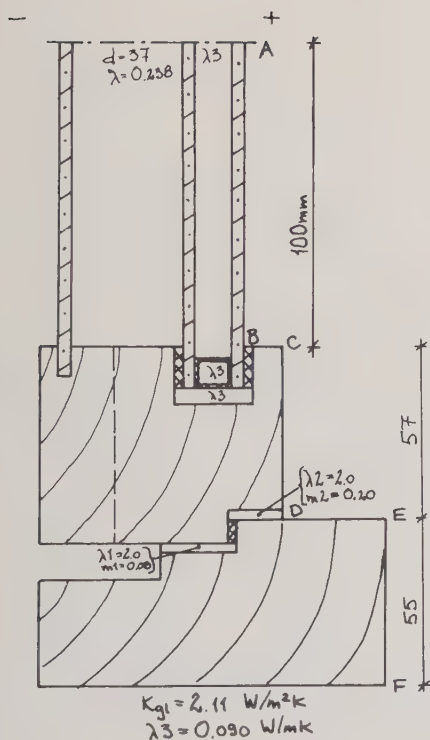
TAB.4.3.17 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of an openable window. Wooden casement with aluminium cladding and wooden frame. $\vartheta_U = -20^\circ\text{C}$, $\vartheta_i = 20^\circ\text{C}$, $m_U = 0.055$ and $m_i = 0.115 \text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of sash according to Case 24-2. Material designations according to TAB. 4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
24-1	AB	7.6	107.0	2.68
	BC	10.6	80.9	2.02
	CD	13.0	86.5	2.16
	DE	14.2	48.5	1.21
	EF	15.9	34.2	0.86
24-2 $m_{BC} = 0.28$	AB	7.4	108.8	2.72
	BC	7.2	45.6	1.14
	CD	12.3	93.1	2.33
	DE	14.2	48.6	1.22
	EF	15.9	34.3	0.86

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^\circ\text{C}$)
	k_{AF}	k_{BF}	k_W	
24-1	2.62	2.56	2.48	-1.6
24-2	2.55	2.40	2.44	-2.7

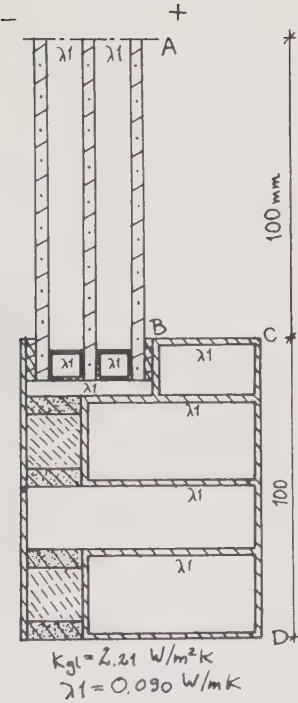
TAB.4.3.18 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of an openable window. Wooden casement and frame. $\vartheta_U = -20^\circ\text{C}$, $\vartheta_i = 20^\circ\text{C}$, $m_U = 0.055$ and $m_i = 0.115 \text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of sash according to Case 25-2. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur- face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
25-1	AB	10.0	86.6	2.17
	BC	12.0	67.1	1.68
	CD	13.4	91.6	1.29
	DE	15.0	42.1	1.05
	EF	15.8	35.4	0.89
25-2 $m_{BC} = 0.29$	AB	9.9	87.7	2.19
	BC	10.3	33.1	0.83
	CD	13.0	94.8	2.37
	DE	15.0	42.2	1.06
	EF	15.8	35.4	0.89

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B ($^\circ\text{C}$)
	k_{AF}	k_{BF}	k_W	
25-1	2.14	2.11	2.13	7.0
25-2	2.12	2.06	2.12	6.3

TAB.4.3.19 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of an openable window. Aluminium casement and frame. $\dot{v}_U = -20\text{ }^{\circ}\text{C}$, $\dot{v}_i = 20\text{ }^{\circ}\text{C}$, $m_U = 0.055$ and $m_i = 0.115\text{ m}^2\text{K/W}$. Theoretically calculated surface resistance for bottom rail of sash according to Case 26-5. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.

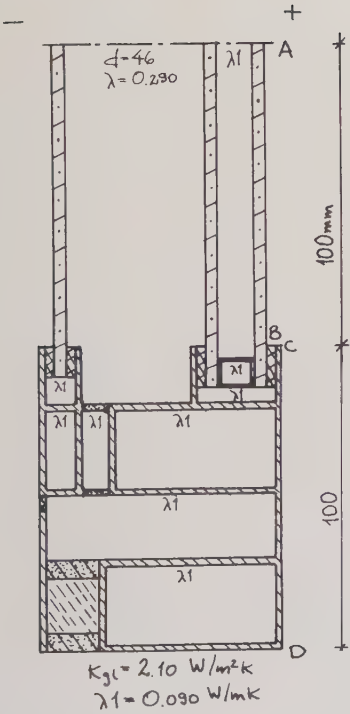


Case	Sur-face	Mean temp ($^{\circ}\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
26-1 $\lambda_1 = 0.090$	AB	9.1	94.0	2.35
	BC	7.8	106.1	2.65
	CD	7.9	105.5	2.64
26-2 $\lambda_1 = 0.12$	AB	9.2	93.8	2.35
	BC	7.4	109.8	2.75
	CD	7.4	109.4	2.74
26-3 $\lambda_1 = 0.060$	AB	9.1	94.4	2.36
	BC	8.2	102.1	2.55
	CD	8.4	101.2	2.53
26-4 $\lambda_1 = 0.030$	AB	9.0	95.0	2.38
	BC	8.9	96.7	2.42
	CD	9.0	95.5	2.39
26-5 $\lambda_1 = 0.090$ $m_{BC} = 0.29$	AB	8.9	95.6	2.39
	BC	6.4	54.3	1.36
	CD	6.6	116.3	2.91

Case	k-value ($\text{W/m}^2\text{K}$)			$\dot{v}_B$ ($^{\circ}\text{C}$)
	k_{AD}	k_{BD}	k_W	
26-1	3.01	3.67	2.72	4.7
26-2	3.08	3.81	2.77	4.8
26-3	2.94	3.52	2.68	4.6
26-4	2.85	3.33	2.62	4.4
26-5	2.91	3.44	2.66	3.8

TAB.4.3.20 Mean temperatures, mean flow rates, k-values and minimum temperatures along the inside of an openable window. Aluminium casement and frame. $\vartheta_u = -20^\circ\text{C}$, $\vartheta_i = 20^\circ\text{C}$, $m_u = 0.055$ and $m_i = 0.115 \text{ m}^2\text{K/W}$.

Theoretically calculated surface resistance for bottom rail of sash according to Case 27-2. Material designations according to TAB.4.2.2. The k-value for a 1x1 m window has been calculated on the basis of the construction in the case concerned around the entire window.



Case	Sur-face	Mean temp ($^\circ\text{C}$)	Mean flow (W/m^2)	spec flow ($\text{W/m}^2\text{K}$)
27-1	AB	10.2	84.3	2.11
	BC	4.8	129.4	3.24
	CD	3.8	141.0	3.53
27-2 $m_{BC}=0.23$	AB	10.2	84.8	2.12
	BC	4.4	67.1	1.68
	CD	3.6	142.4	3.56

Case	k-value ($\text{W/m}^2\text{K}$)			ϑ_B	ϑ_{C-D}
	k_{AD}	k_{BD}	k_W	($^\circ\text{C}$)	($^\circ\text{C}$)
27-1	2.90	3.69	2.61	6.0	3.8
27-2	2.88	3.64	2.60	5.7	3.6



FIG.4.3.1
Temperature profile along the inside of the window. Cases 1-1, 1-2, 1-4, 1-6 and 1-7 (see TAB.4.3.1).

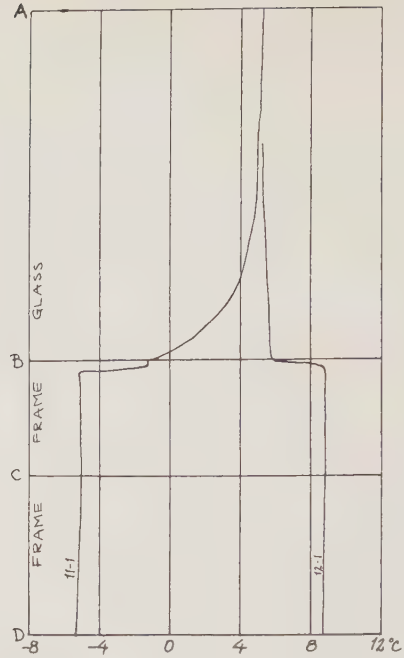


FIG.4.3.2
Temperature profile along the inside of the window. Cases 11-1, 12-1 (see TAB.4.3.6 and 4.3.7).

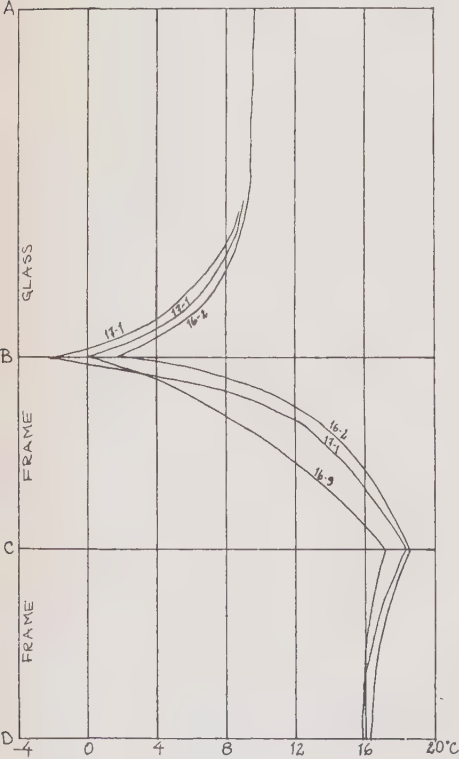


FIG.4.3.3
Temperature profile along the inside of the window. Cases 16-2, 16-9 and 17-1 (see TAB.4.3.9-10).

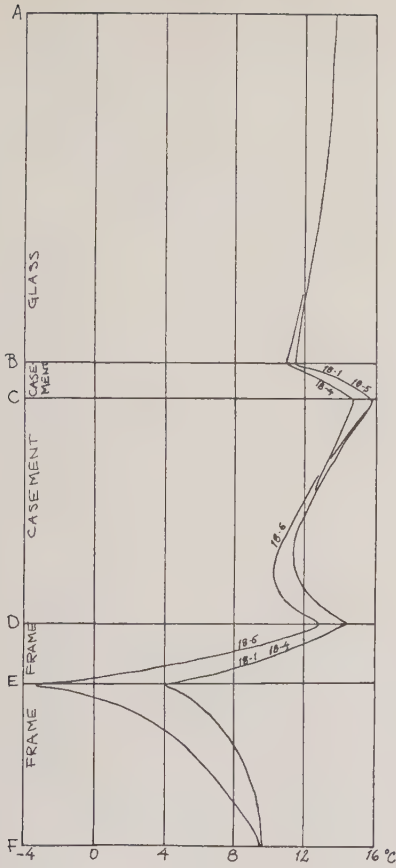


FIG.4.3.4
Temperature profile along the inside
of the window. Cases 18-1, 18-4 and
18-5 (see TAB.4.3.11).

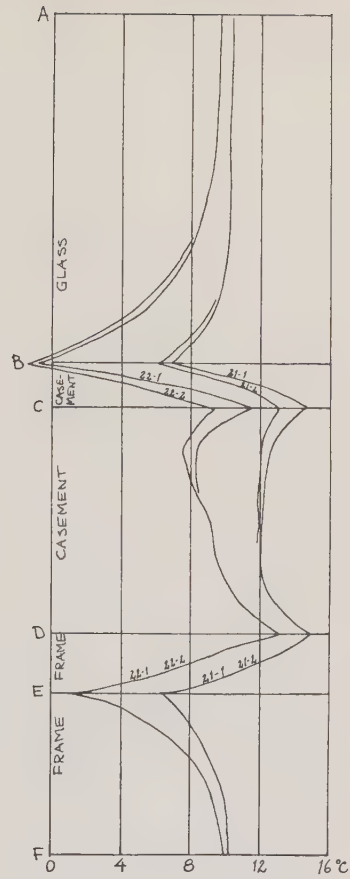


FIG.4.3.5
Temperature profile along the inside
of the window. Cases 21-1, 21-2,
22-1, 22-2 (see TAB.4.3.14 and 4.3.1

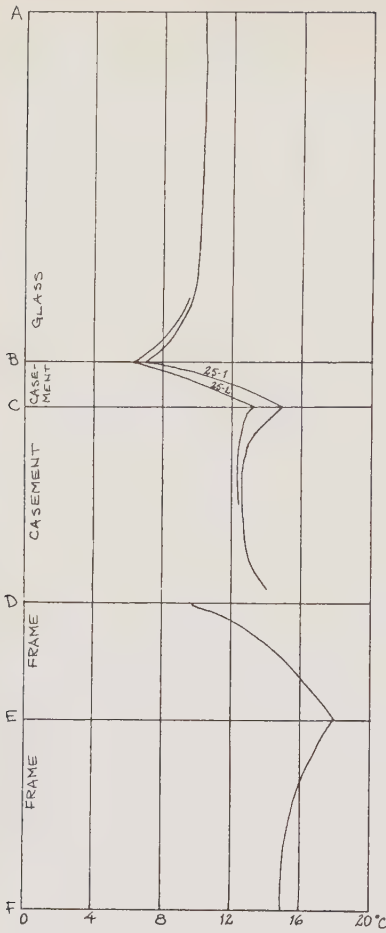


FIG.4.3.6

Temperature profile along the inside of the window. Cases 25-1 and 25-2 (see TAB.4.3.18).

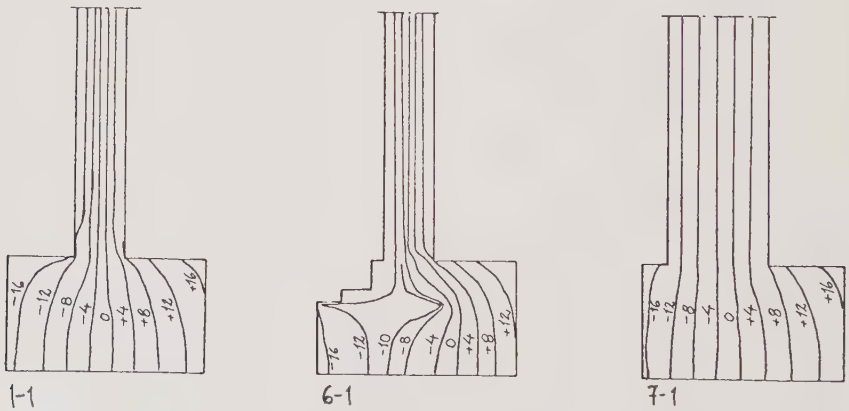


FIG.4.3.7-4.3.9 Isotherms for cases 1-1, 6-1 and 7-1. See TAB.4.3.1, 4.3.3 and 4.3.4. $t_U = -20^\circ\text{C}$, $t_i = 20^\circ\text{C}$, $m_U = 0.055 \text{ m}^2\text{K/W}$ and $m_i = 0.115 \text{ m}^2\text{K/W}$

5 METHODS OF MEASUREMENT FOR TEMPERATURES AND HEAT FLOWS

5.1 HOT BOX

In order that heat transfer through a construction may be studied a test arrangement has been designed. In this test arrangement the thermal characteristics of different windows or wall constructions can be studied. The equipment comprises three parts: a cold room, a warm room and a measuring box. In the literature this box is often referred to as the hot box or guarded hot box. The test window to be studied is mounted in a special wall element. This element is placed between the cold and warm rooms. The portable five sided measuring box which is mounted over the test specimen is also situated in the warm room. The test specimen thus constitutes the sixth side of the box, and it is surrounded by a tightly fitting outer shell. Both the measuring box and the warm room contain a heat source which creates a uniform temperature in both these spaces. An endeavour is made to maintain the temperature difference across the walls of the measuring box at a minimum value. This means that the energy supplied to the measuring box mainly passes through the test specimen and the wall element surrounding this.

By using this method to measure the energy supplied to a measuring box,

- * the mean flow over a large area is measured and the thermal transmittance is obtained for the entire area
- * the method can be applied for multidimensional flow
- * the surface temperatures need not be uniform within the construction

On the other hand, these measurements provide

- * no information concerning the distribution of heat transfer over the surface or within the construction
- * a convection pattern inside the measuring box and a

convective surface coefficient of heat transfer
which are usually not the same as those in a building

For a window exposed to the wind and weather, the boundary conditions vary all the time, and there is an infinite number of variations. On the inside, an influence is exerted by the geometric configuration of the window and adjacent parts, the heating and ventilation system, the design of the room, the convective air flow and the radiant heat transfer between surfaces, i.e. the boundary conditions on the inside of the window. It is also difficult to define a normal state or some kind of mean value, since these depend to a great extent on the purpose for which the test results are to be used (e.g. the design of a heating installation, determination of indoor temperatures in the summer).

Another factor which must be considered is repeatability, i.e. the possibility of repeating the experiment either in the same test arrangement or in a similar arrangement constructed elsewhere. In such a case it is essential that the boundary conditions over the construction are the same, i.e. it must be possible for the parameters which determine the boundary conditions to be measured and controlled.

5.1.1 Cold room

The construction of the test arrangement is illustrated in FIG.5.1.1. The cold room has a floor surface 1.12x3.50 m, with the distance between floor and roof being 2.52 m. A ceiling of perforated wood fibre board reduces this height, so that a gap of 0.29 m is formed below the roof. The walls, floor and roof are made of a timber stud construction with intermediate mineral wool insulation of about 120 mm thickness. The wall element between the cold and warm rooms consists of two parts made of 200 mm slabs of expanded polyurethane sandwiched between 10 mm sheets of plywood, with a through 10 mm plywood only at the free edges. The outer part comprises an element of 2.4x2.4 m overall dimensions, with a central opening of 1.9x1.9 m. Another wall element of 1.9x1.9 m overall dimensions, with a 1200x1200 mm window opening,

is placed inside this opening. If the window opening has to be made smaller, make-up pieces of the same general construction as the wall element are used. The refrigeration equipment consists of an air cooled compressor of 1.5 kW rating and a condenser. A capillary tube thermostat controls the temperature by stopping/starting the flow of gas between the compressor and condenser. The pressure in the gas pipe is controlled by a pressurestat. The condenser is placed near the roof along one of the short sides in the cold room. Two fans draw the air current past the cooling fins and further into the gap below the roof. The cold air is forced through the perforated board and down towards the floor. In the cold room there are 4 small blowers which create a horizontal air flow of about 1 m/s perpendicular to the test specimen. In this way a uniform air temperature is obtained at the specimen and similar boundary conditions over the cold surface. The blowers are situated 700 mm in front of the cold side of the wall element and are placed symmetrically in front of the window opening.

Since the air temperature is controlled by on/off regulation of the compressor, the air temperature in the cold room varies with time. This means that the air temperature oscillates about the mean value ($\bar{T}_U$) with an amplitude of 0.5 °C and a period varying between 90 s ($\bar{T}_U = -5$ °C) and 140 s ($\bar{T}_U = -25$ °C).

The surface temperature however varies much less, on a pane of glass the amplitude is 0.1-0.15 °C. Uniformity of air temperature in front of the window opening has been checked by measurements in the air current and measurements on a sheet of insulating material inserted into the window opening. The temperature sensors have also been calibrated in relation to one another. The difference between the highest and lowest value has been found to be about 0.1 °C.

5.1.2 Warm room

The warm room is separated from the laboratory by a construction which is partially insulated. The floor area is 2.7x4.4 m and the height 2.9 m. The room is heated by an electric radiator of 1600 W maximum output situated right in front of the measuring box. In

order to prevent direct radiation between the measuring box and radiator, a screen is placed in front of the radiator. Temperature is controlled by an electronic thermostat of $\pm 0.1^{\circ}\text{C}$ accuracy. The temperature in the warm room is maintained about 2°C higher than in the laboratory.

5.1.3 Measuring box

The measuring box has five closed sides (FIG.5.1.2). It consists of 100 mm slabs of expanded polyurethane sandwiched between 8 mm sheets of plywood, without through studs. The internal dimensions (width $\times$ height $\times$ depth) are 1697 $\times$ 1697 $\times$ 610 mm. The box is portable and can be placed over the wall element in different ways. However, the box has so far always been placed symmetrically about the window opening. An 80 mm wide strip of plastics coated glass fibre wool which is placed around the open side of the measuring box has been used to provide a seal between the box and the wall element. When the measuring box is mounted on the wall element by means of four fixtures, the plastics coated strips are compressed to a certain extent. The measuring box is heated by an electric tubular heater of 345 W rating placed near the bottom parallel to the wall and floor. The heater is controlled by a thermostat placed in the middle of the box. In order to control natural convection inside the box and to prevent direct radiation from the heater towards the window, a vertical screen is installed in the middle of the box. Air circulates through spaces above and below the screen. The temperature difference over the walls of the box is measured by eight series-coupled copper-constantan thermocouples, four placed over the rear wall and the remaining four one each in the middle of the other four walls. The air temperature sensor is placed near the opening in the measuring box, approximately 100 mm from the test window.

Circulation of air in the measuring box is maintained by natural convection, and there is therefore a temperature gradient in the vertical direction in the box. This gradient is due to losses in the box, mainly fabric losses through the test window. The relationship governing this loss has been found to be approximately

$$\Delta T = 0.07 P^{0.7} \quad (5.1)$$

where

ΔT = temperature gradient ($^{\circ}\text{C}/\text{m}$)
 P = heat output (W)

A heat output of 85 W thus gives rise to a temperature gradient of $1.6^{\circ}\text{C}/\text{m}$.

5.1.4 Measuring procedure

Measuring equipment comprises a Doric Digitrend 220 datalogger with 79 channels for copper-constantan thermocouples or voltage measurement, and 10 channels for Type Pt100 resistance transducers. Temperatures are measured with a resolution of 0.1 K and the rate of reading is 2 channels/s. The measuring system has been supplemented with a Hewlett-Packard HP85 desk top computer of 32k memory. This computer controls collection of readings and also exercises a certain measure of control over data. It is also possible to check readings between successive measurements. Generally speaking, measurements have been made two to four times per hour. In this way a clear picture is obtained of steady conditions during measurements. In addition it is possible for a limited number of channels to be read more often between the main measurements. During the main measurements mean values are formed for these channels, and it is obviously also possible to calculate and record other statistical parameters such as standard deviation, minimum/maximum values. It is primarily the values which vary between measurements which are read, such as the air temperature in the cold room, the measuring box and the warm room.

The temperatures in the three spaces (cold room, measuring box and warm room) are controlled individually by thermostats which are set manually. The temperature levels in the box and the warm room are not altered during the measurements, but are set in such a way as to result in minimum net heat transfer through the walls of the measuring box.

The instantaneous power supplied to the measuring box is changed by a measuring transducer into voltage. This voltage is then integrated in a pulse integrator with a 0-10 V output signal.

The temperature difference over the walls of the measuring box, as measured by the eight series coupled thermocouples, is read off as a voltage in μV . This voltage is converted into temperature by means of a second degree curve.

When measurements begin, it is 2-4 hours before the air temperature in the cold room has attained the preset value. Steady conditions in the measuring box are obtained after another 4 hours. When evaluations are carried out, the variations in air temperature and the power supplied to the box are studied. The measuring period during which variations in power supply are a minimum is determined. As a rule, the standard deviation is between 1 and 2.5 W for a total power of approximately 100 W, depending on conditions during measurements. The difference between the maximum and minimum value is about four times the standard deviation. However, the mean values for each hour differ from one another by less than 1%.

5.1.5 Calibration of measuring box

For steady conditions, the heat balance for the measuring box is

$$P_B = k_{cs} A_{cs} (T_i - T_u) + k_v A_v (T_i - T_u) + k_B A_B (T_i - T_R) + n_u V \rho c_p (T_i - T_u) + n_R V \rho c_p (T_i - T_R) + R \quad (5.2)$$

P_B = heat input supplied in the measuring box (W)

k_{cs}, k_v, k_B = thermal transmittance for the calibration slab, wall and measuring box respectively ($\text{W/m}^2\text{K}$)

A_{cs}, A_v, A_B = areas (m^2) of calibration slab, wall and measuring box respectively

T_i, T_u, T_R = temperatures (K) in the measuring box, cold room and warm room respectively

n_u, n_R = air change rate (1/s) due to the air current to and from the cold room and warm room respectively

- V = volume of measuring box (m^3)
 c_p, ρ = specific heat capacity and density
 ($\rho c_p = 1220 \text{ J/m}^3\text{K}$ for air at 20°C)
 R = residual term representing losses at the junctions
 between measuring box and wall and between wall and
 window (calibration slab) due to the temperatures
 T_i , T_u and T_R and the adjoining materials.

In order to determine the airtightness of the measuring box, the number of air changes was measured during a measuring period when the temperature difference between measuring box and cold room was approx. 39°C ($T_i \approx 24^\circ\text{C}$, $T_u \approx -15^\circ\text{C}$). The seal around the measuring box consisted only of the plastics coated glass fibre strip, and all cable connections through the measuring box and all joints in the test window were taped up. The air change rate was found to be very low, approximately 0.02 per hour.

In order to find what voltage in the eight series coupled thermocouples corresponds to zero temperature difference, a thermopile was constructed. Calibration was carried out by immersing all thermocouples in a bath of water. It was found after a number of measurements at different water temperatures that 0°C was represented by a voltage of $0 \pm 2 \mu\text{V}$. It is reasonable to assume that this value remains the same after the thermocouples are affixed to the walls of the measuring box. Since each pair of thermocouples in the thermopile is attached to surfaces of about the same area, each pair has the same weighting and a mean temperature difference is thus obtained over the measuring box. A voltage of $0 \mu\text{V}$ across the thermopile thus implies that there is zero net heat exchange between the measuring box and the warm room. Reservations must naturally be made with regard to the representation by each pair of thermocouples of the mean temperature over their area, since temperature lows or highs may obviously occur over an area. However, nothing was found that may indicate that such is the case.

Two different calibration elements have been used for calibration of the measuring box. In principle, the first element (K1) is constructed in the same way as the wall element, i.e. 200 mm

expanded polyurethane slabs sandwiched between 10 mm sheets of plywood. The second calibration element (K2) is a sheet of expanded polyurethane 20.5 mm thick. Both elements have overall dimensions of 1200x1200 mm, and they have been placed flush with the inside of the wall element. Since the convection pattern in the measuring box changes as the power supplied increases, for the two calibration elements the same temperature difference across the measuring box will not give rise to the same change in power. At a temperature difference of 45 K, the power supplied for the calibration elements K1 and K2 is 10.5 W and 67.5 W respectively. The comparable measuring case for a temperature difference of 20 K is a normal window construction for which the power is 66 W. During calibration, the temperature difference for the thermopile between the box and the warm room has been varied about -4°C and $+2.5^{\circ}\text{C}$. An endeavour has been made during all measurements to keep the temperature difference between measuring box and cold room at 45 K, and for this reason the transmission of heat through the wall element is the same in all tests with the same calibration slab. With $n_U = n_R = 0$, the heat balance equation is

$$\begin{aligned} P_B &= (T_i - T_U)(k_{CS}A_{CS} + k_VA_V) + k_BA_B(T_i - T_R) \\ &= C_1(T_i - T_U) + C_2(T_i - T_R) \end{aligned} \quad (5.3)$$

Since the temperature difference between the box and the cold room ($T_i - T_U$) has been maintained constant, the difference in the loss through the walls of the box can be determined by means of the relationship $C_2(T_i - T_R)$.

In FIG.5.1.3 the power is plotted as a function of the temperature difference between measuring box and warm room. The method of least squares has been used to determine the regression lines, $P_B = 24.7 - 3.82 \Delta T$ ($R = -0.997$) for K1 and $P_B = 95.7 - 4.12 \Delta T$ ($R = -0.996$) for K2, where $\Delta T = T_i - T_R$. The measuring box thus has a specific fabric loss of 3.82 W/K and 4.12 W/K respectively.

For element K1 with $k_{K1} = 0.162 \text{ W/m}^2\text{K}$, $k_{CS} = k_V$, and at $T_i - T_R = 0$ the loss through the wall and calibration slab K1 is

$$0.162(A_{CS} + A_V)45$$

Since placing of the sealing strip is displaced towards the outside surface of the box, the area should be about $3.24\text{--}3.42\text{ m}^2$ (lengths of sides $1.80\text{--}1.85\text{ m}$), which yields a loss of $23.6\text{--}24.9\text{ W/K}$. This is to be compared with the measured value of 24.7 W/K .

For the surrounding wall element only we then have $0.162(1.84^2 - 1.2^2) = 0.32\text{ W/K}$. To this must be added extra losses through the wall due to three dimensional flow at the corners of the window opening.

5.1.6 Calibration of the wall element

The object has been to determine the fabric losses through the wall between the cold room and the hot box. The total losses from the box comprise losses through the test specimen, the wall and the hot box. The last of these has been studied above. If the test specimen consists of some well known material of known properties, the total fabric losses through the test specimen can be calculated. The remaining losses are then obtained as a residual item, the total losses less the losses through the hot box and test specimen.

Test window P1-P7 with their different frame constructions have been calibrated with a Gullfiber 1275 facade sheet of 50 mm or 100 mm thickness, surrounded by 3.2 mm hardboard or $3\text{--}4\text{ mm}$ glass panes.

Determination of λ for the facade sheet has been carried out at the Gullfiber laboratory (1983-03-22) by means of Lang measurements according to Swedish Standards SS 02 42 11. Measurements have been made on three specimens of 50 mm thickness and three of 100 mm thickness. In addition the outer layer was removed from the three 100 mm thick specimens, making the thickness 96 mm . This removal of the surface layer did not cause any significant change in the value of λ . For all nine samples the value of λ_{10} is $0.0307 \pm 0.0005\text{ W/mK}$, the densities of the nine samples being $53\text{--}60\text{ kg/m}^3$. No direct relationship appears to exist between λ and density. This is not really surprising, since λ has a minimum value for a density of approximately $50\text{--}70\text{ kg/m}^3$, and the curve

is quite flat. The thermal resistance for the 50 mm thickness was $m=1.63 \text{ m}^2\text{K/W}$, and for the 100 mm thickness, $m=3.26 \text{ m}^2\text{K/W}$.

To start with, sheets of 3.2 mm hardboard were glued to the facade sheet. The variation in the results was however greater than desirable, presumably due to deflection of the hardboard. This gives rise to differences in the thickness of the test specimen, causing variations in thermal resistance. A stiffer sheet of glass was then used, the insulation being clamped between the sheets of glass with strips of wood. The different frame configurations for the test windows are illustrated in FIG.6.1.1-6.1.3. For test windows P1, P2, P3 and P4 the frame construction was changed by removing the wooden spacer between the sheets of glass. On the cold side the pane of glass was held fixed by 10x20 mm strips of wood. The insulation slab was 5-7 mm smaller than the dimension between glazing rebates, see TAB.5.1.1.

Losses through the test specimen have been calculated as follows. According to the calibration, the Gullfiber 1275 facade sheet has a value of $\lambda=0.0307\pm 0.0005 \text{ W/mK}$ at a mean temperature of $+10^\circ\text{C}$. If the temperature dependence is $0.00012 \text{ W/mK per } ^\circ\text{C}$ (according to the Gullfiber product description), then

$$\lambda = 0.0295 + 0.00012 \bar{\vartheta}_m \text{ (W/mK)}$$

where $\bar{\vartheta}_m$ = mean temperature across the specimen.

On the surface sheets there are five temperature sensors (copper-constantan thermocouples) on the inside and three on the outside. In this way the mean surface temperatures on the inside and outside, and thus the mean temperature across the insulation, can be calculated.

Calculation of the surface resistance is based on the ratio of the difference in temperature between the surface and air to the difference in temperature across the test specimen.

$$m_i = \frac{\vartheta_i - \vartheta_{is}}{\vartheta_{is} - \vartheta_{us}} m$$

$$m_u = \frac{t_{us} - t_u}{t_{is} - t_{us}} m$$

where

- t_i, t_u = air temperature inside and outside ($^{\circ}\text{C}$)
 t_{is}, t_{us} = surface temperature on the inside and outside ($^{\circ}\text{C}$)
 m = surface resistance of the test specimen (insulation + surface sheets)

The thermal transmittance (k) of the test specimen can now be calculated. When this is multiplied by the area of insulation (A), the fabric loss (kA) for the test specimen is obtained.

When the total power is corrected for the power lost through the wall of the hot box and divided by the difference in air temperature ($t_i - t_u$), the fabric loss through the wall and test specimen is obtained. If the fabric loss through the specimen, given above, is deducted from this the fabric loss (T_v) through the wall is obtained.

Insulation of 50 mm thickness has been used for the test windows, and insulation of 100 mm thickness for the window opening alone.

The thermal conductivity of the thermal insulation material in the wall (polyurethane) is assumed to be given by the expression $\lambda = 0.025 + 0.00008 t_m$, where t_m = mean temperature in the wall. The fabric loss through the wall is given by $T_v = Ak$ (W/K), i.e. $T_v = A\lambda/d$ if the surface resistance is ignored. We thus have $T_v = A(0.025 + 0.00008 t_m)/d$. It is thus possible to express the loss through the wall as a linear function of the form $T_v = a + b t_m$. According to measurements set out in TAB.5.1.1, the constant a at $t_m = 0^{\circ}\text{C}$ is approximately 0.7 for 50 mm insulation and 0.5 for 100 mm insulation. The ratio of the constants a and b must be $0.025/0.00008$, and the value of b should thus be $0.0016 - 0.00224$ W/K per $^{\circ}\text{C}$. It is assumed that $b = 0.0020$ W/K per $^{\circ}\text{C}$ for all test windows, and that $b = 0.0015$ W/K per $^{\circ}\text{C}$ for 100 mm insulation. The effect due to the frame is obtained as the difference between measurements with and without a frame. For

instance, for the 1200x1200 mm window opening the value obtained is $(0.637 + 0.020 \frac{1}{m}) - (0.436 + 0.0015 \frac{1}{m}) = 0.201 + 0.0005 \frac{1}{m}$. When this value is divided by the area of the frame, the thermal transmittance k obtained is approximately $1.7 \text{ W/m}^2\text{K}$ at $\frac{1}{m} = 0$ °C. For the configurations which cannot be calculated in this way (test windows P3 and P4), the k -value is assumed to be $1.7 \text{ W/m}^2\text{K}$, which enables the fabric loss through the frame to be calculated. Test window P5 is almost identical to test window P2, but the wooden frame is replaced by one of expanded polyurethane. Since the thermal transmittance of the wooden frame is approximately $1.65 \text{ W/m}^2\text{K}$, the wood itself has a resistance of about $0.41 \text{ m}^2\text{K/W}$. If the expanded polyurethane frame has a thermal conductivity of approx. $\lambda = 0.027 - 0.040 \text{ W/mK}$, the resistance of the frame is approximately $1.4 - 2.1 \text{ m}^2\text{K/W}$, i.e. a k -value of $0.43 - 0.62 \text{ W/m}^2\text{K}$. Since heat flow is two or three dimensional, it is difficult to obtain correct results by calculation. The distribution of heat flow is also changed when wood is replaced by expanded polyurethane. It is assumed that the thermal transmittance is $0.52 \text{ W/m}^2\text{K}$.

According to these calibrations, the losses from the measuring box in conjunction with measurements on a window of 918x1200 mm overall size are as set out in FIG.5.1.4.

5.2 HEAT FLOW BETWEEN WALL AND FRAME

In order to determine the way in which the thickness of a glass assembly or the thickness of an insulation slab affects heat flows through adjacent components such as the frame and wall, calculations have been carried out in the computer. Two typical details have been studied (FIG.5.2.1), one in which the insulation slab is placed directly into the window opening (Alt 1), and another in which a glass assembly (5 mm glass + air space + 5 mm glass) (Alt 2a) or an insulation slab (Alt 2b), both in a frame, is placed into the window opening. When the slab is placed in a frame, the losses through the wall are practically independent of thickness. If, however, the insulation material is placed into the window opening direct, losses decrease as the thickness of insulation increases. Losses through the frame are considerably

reduced when the thickness of the glass assembly increases. The thermal transmittance of the frame changes from 1.96 to 1.42 $\text{W/m}^2\text{K}$ when the air space is increased from 20 mm to 120 mm. In FIG.5.2.2 the increase in heat flow for the constructions in FIG. 5.2.1, compared with unidimensional flow, has been calculated per metre of wall. The additional flow for a thickness of 0.02 m is 0.05–0.07 W/mK , while for thicknesses greater than about 0.1 m the additional flow for the insulation slabs is less than 0.01 W/mK . For the air spaces, even for relatively wide ones, the additional flow is relatively high. A change in the width of air space from 10 mm to 50 mm gives rise to a change in heat flow from 0.07 W/mK to 0.0455 W/mK . For the format 1200x1200 mm and $\Delta\vartheta=40\text{ K}$, this means that the power is changed from 13.44 W to 8.74 W, and the thermal transmittance is changed by $4.7/(1.2 \cdot 1.2 \cdot 40) = 0.08\text{ W/m}^2\text{K}$.

It is seen from FIG.5.2.2 that for a 100 mm thick insulation slab according to Alt 1 the additional flow is about 0.013 W/mK , while for a 50 mm insulation slab according to Alt 2b it is about 0.030 W/mK . For the 1.2x1.2 m window opening this gives 0.065 W/K and 0.15 W/K respectively. According to calculations for unidimensional conditions, the wall has a specific flow of $0.16(18.5^2 - 1.2^2) = 0.32\text{ W/K}$, the corresponding value for the frame being $0.90 \cdot 4(1.2 - 0.039)0.039 = 0.16\text{ W/K}$. It can thus be estimated that the fabric loss through wall and frame, for a 50 mm insulation slab according to Alt 2b, is $0.32 + 0.16 + 0.15 = 0.63\text{ W/K}$, and the fabric loss through the wall for a 100 mm insulation slab according to Alt 1 is $0.32 + 0.065 = 0.39\text{ W/K}$. This is to be compared with the measured values of 0.637 W/K and 0.436 W/K according to the calibration tests in TAB.5.1.1. The difference is not particularly large.

In FIG.5.2.3 the calculations have been summarized for a window construction installed in a 1200x1200 mm window opening. It can be seen that the energy supplied to the box decreases as the width of the space increases. In the case of spaces larger than 0.05 m this is due to the reduction in losses through frame and wall. This makes it difficult for the resistance of the air space to be defined easily without the necessity to carry out long and comprehensive calculations for the influence due to frame and

wall.

In Chapter 6 which sets out the experimental measurements it has not been considered feasible to apply such a method to correct for the influence due to frame and wall. By means of calibration tests, a value for the fabric loss through wall+frame, and another for the fabric loss through the frame, has been determined for an air space of 50 mm. According to FIG.5.2.3, the loss through wall+frame varies from 0.83 to 0.59 W/K, i.e. by $\pm 3-4\%$ of the energy supplied to the air space (of 50 mm width). This has some significance for the determination of the air space resistance. If, however, we study the typical curve setting out the relationship between thermal resistance and width of air space, we see that it has some significance for determination of the maximum thermal resistance. This can be seen in FIG.6.3.12 where, after correction, the maximum appears to be situated between 80 and 150 mm.

5.3 TEMPERATURE MEASUREMENTS

5.3.1 Thermocouples

When measurements are made with thermocouples, one junction is placed at a measuring point and the other at a reference point. A bath of thermostatically controlled temperature (mostly 0°C) is used as the reference temperature. It is also possible to use a reference of variable temperature, in which case the temperature of the reference is determined at the time of each measurement by means of thermistors or resistance transducers. In order that an accurate temperature may be obtained, it is essential that the temperature of the reference is well defined.

Generally speaking, tables for the accuracy of thermocouples quote comparatively large deviations from the basic values. For copper-constantan, DIN specifies a permissible deviation of 3°C for measurements at -100°C and $+100^{\circ}\text{C}$. The accuracy of the voltmeter and the reference bath must be added to this figure. Accuracy may however be improved by calibrating the sensors and checking the measuring system as a whole. As regards calibra-

tion of sensors, a distinction must be made between comparative and absolute calibration. In absolute calibration the temperature is compared with a reference, which demands sophisticated equipment. In comparative calibration a number of sensors are compared with one another. In this way temperature differences can be measured with greater precision. The relationship between thermal emf (U) and temperature (ϑ) can be simplified to a linear one of the form $U=a+b\vartheta$, where b is the slope of the line and a the thermal emf at 0°C . In comparative calibration at a certain temperature, the curves for all thermocouples are displaced laterally, i.e. $U=a+b\vartheta-c$, where c is the lateral displacement. By means of comparative calibration at two or more temperature levels the slope b of the curve can be checked. In order that an idea of the absolute level may be obtained, an accurate reference bath can be used and the constant a can then be determined with good accuracy.

From a Type T thermocouple roll (copper-constantan), 40 thermocouples were made. Data collection consisted of a voltmeter, scanner, reference bath and a control unit. All the thermocouples were placed in a bath containing silicon oil which was kept in vigorous circulation by means of propellers. The junctions of the thermocouples had been wrapped in plastics foil to prevent contact between these. They were then packed in lots of five in plastics bags. The thermocouples were then immersed to the greatest possible depth in the bath. In this way thermal conduction through the thermocouple wires is a minimum, and the temperature of the junction assumes the temperature of the silicon bath. The temperature of the bath was set at six different levels, approximately $+1.3$, $+4.9$, $+9.3$, $+14.7$, $+20.3$ and $+30.1^{\circ}\text{C}$. Readings were taken every five minutes.

The difference between the highest and lowest value ($\vartheta_{\max} - \vartheta_{\min}$) varied between 0.14 and 0.40, and the standard deviation between 0.03 and 0.10 for the six temperature levels. The mean for all six cases is $\vartheta_{\max} - \vartheta_{\min} = 0.24$ and $s = 0.06$, i.e. 67% of the values are situated within ± 0.06 from the mean value.

In addition, 10 surface sensors of Type T thermocouple (copper-

constantan) were compared according to the same method. The standard deviation and the difference between the highest and lowest value was less than for the thermocouple wire. Five thermocouples were also calibrated at the same time for use as reference. The mean difference between the different types, surface sensors and thermocouple wires, was less than 0.03°C .

The conclusion that can be drawn is that very accurate measurements can be made with thermocouples. If the thermocouples are made from the same roll, differences between thermocouples will be small. If accurate measurements are to be made, calibration must be carried out. In many cases comparative calibration is sufficient.

5.3.2 Measurement of surface temperatures

When the thermal characteristics of windows are being determined, it is essential that surface temperatures should be measured with great accuracy and reliability. Since a window generally has a high thermal transmittance, the external surface resistance influences the total thermal resistance. This means that there are large temperature differences, particularly between indoor air and the inside surface of the glass. In addition, there are large temperature gradients inside the air space in the window.

If a surface temperature sensor is not in proper contact with the surface, ambient air will affect the measured value of the surface temperature. The value will be influenced by air temperature. Since the thermal resistance of a double glazed window is of the same order as the sum of the outer and inner surface resistances, this means that one half of the temperature difference between indoor and outdoor air occurs across the window construction. For a window where the difference between the temperatures of indoor and outdoor air is 20°C , about 10°C occurs across the window construction and 10°C at the surfaces. If there is a 5% error in the temperature difference across the construction, i.e. 0.5°C , this means that the thermal resistance calculated across the construction will also be 5% in error. The error in thermal resistance between indoor and out-

door air will then be about 2.5%.

In order that an idea may be obtained of the way in which different methods of attachment affect results, a study was carried out. The sensors were attached to a glass surface (the outside of an aquarium). Inside the aquarium there was either a water-ice mixture which had a constant temperature of approximately 0 °C, or pure water at different temperature levels ranging from five to thirty degrees. It was not possible to maintain these temperatures absolutely constant, and they were allowed to drift during measurements. The room temperature was maintained at about 19-20 °C. Care was taken during measurements to ensure that the water-ice mixture or the water bath was kept in vigorous circulation so that the temperature in the bath may be uniform. The sensors were attached by different techniques to the glass surfaces over an area of 60x80 mm. Readings were taken at intervals of 1 or 2 minutes.

In evaluating the results, the "true" temperature of the outside of the glass surface was calculated. A mean value of the ambient air temperature and the water bath was first calculated. The inner surface coefficient of heat transfer was calculated by the formulae

$$\alpha_C = 2.2(T_i - T_V)^{1/4} \quad (5.4)$$

$$\alpha_R = \epsilon_{RES} \sigma (T_i^4 - T_V^4) / (T_i - T_V) \quad (5.5)$$

where

T_i, T_V = temperature of air and water bath respectively (K)
 σ = Stefan-Boltzmanns konstant ($5.67 \cdot 10^{-8} \text{ W/m}^2 \text{ K}^4$)

$$\frac{1}{\epsilon_{RES}} = \frac{1}{\epsilon_1} + \frac{A_1}{A_2} \left(\frac{1}{\epsilon_2} - 1 \right), \text{ for a small area surrounded by a large area, } \epsilon_{RES} \approx \epsilon_1$$

In calculating the contribution due to radiation, it was assumed that the temperature of the surrounding surfaces was the same as the air temperature.

For numerical cases, $\epsilon_1=0.8$ was used. For aluminium tape, $\epsilon_1=0.2$ was used. If ϵ_1 is changed from 0.2 to 0, there is a very moderate change in the value of α_i , the calculated surface temperature drops by 0.12°C which is 0.6% of the total temperature difference between the indoor air and the water bath.

The surface coefficient of heat transfer (α_i) is the sum of the contributions due to convection and radiation, i.e.

$\alpha_i = \alpha_c + \alpha_r$. The temperature difference across the glass (Δt_{gl}) is calculated as

$$\Delta t_{gl} = \frac{m_{gl}}{m_{gl} + m_i} (t_i - t_v)$$

where m_{gl} = thermal resistance across the glass

$$m_i = 1/\alpha_i$$

t_i, t_v = air temperature and water temperature

The temperature of the glass surface (t_{is}) is then

$$t_{is} = \Delta t_{gl} + t_v$$

TAB.5.3.1 summarizes these cases. It sets out the differences between measured and calculated values as the percentage of the temperature difference between air and water bath. The thermocouple wires are of the N/N-24-T type, i.e. copper-constantan wound with nylon. The surface sensor is Type Chino C060-CC (copper-constantan). The thickness of the sensor is 0.05 mm, and at the junction the metal is surrounded by a 0.2 mm thick plastics disc.

As will be seen from the table, different kinds of tape have been used, masking tape, woven tape and double sided tape. The error varied between 7% and 19%. Removal of the outer nylon skin from the thermocouple wire, taping of the entire length of the wire or attachment in the form of a spiral had no appreciable effect on accuracy.

When the thermocouple wire was glued to the glass, accuracy was

considerably improved. The figures indicate that the type of adhesive is of secondary importance, the main significance of the adhesive is that it properly fills up the space between the surface and the thermocouple wire. Of the different types of adhesive, school adhesive is water soluble, takes a long time to dry and is impractical in use. Contact adhesive takes a long time to apply. In spite of the rather laborious mixing of the two-pack adhesives Plastic Padding and Araldite, these are preferable. They fill the joint properly, have a suitable consistency, stiffen without allowing ingress of air and have good adhesion with the glass surface which does not deteriorate during the measurements. In spite of the fact that isolated readings deviate from the large bulk of readings, no difference can be detected due to wound or unwound wires or to differences in the length of wire glued to the surface. Large errors may be introduced if the length of wire glued to the surface is too short, but a length of 40-50 mm should be sufficient.

Application of silicon paste with high thermal conductance between the surface and the thermocouple wire gives quite good results. The difficulty is keeping the silicon paste in place below the wire so that it does not spread and reduces adhesion of the tape.

Attachment with aluminium tapes shows very small differences from the calculated surface temperature. As a matter of principle, however, it is doubtful whether aluminium tape should be used as the radiant heat transfer for the actual surface and the taped sensor will be different.

The surface sensors exhibit smaller errors than the thermocouple wires. For tape the error is between 2% and 5%, with a mean of 3.9%. For the double sided tape the error is a little less, the mean value being 2.2%. However, there is a lot of variation in the results when the double sided tape is used, it is obviously quite easy for the sensor to become detached from the surface and lose direct contact with this. The best results are obtained if the sensor is attached using silicon paste, the mean error in this case is about 1.8%.

All the thermocouple wires have been taken from the same roll, and there is therefore little scatter between thermocouples. The surface sensors are also from the same consignment and differences between these are small. For both the thermocouple wires and sensors, calibration tests have been carried out, and these confirm the above (see Subsection 5.3.1). In order that comparison between sensors may be satisfactory, a correction has been introduced on the basis of the calibration tests. If a sensor deviates by more than 0.02°C from the mean value obtained during calibration, this correction has been applied in the table. Any further correction is not meaningful.

5.4 MEASUREMENT OF HEAT FLOW

5.4.1 Temperature difference

For certain special windows measurements were made after the heat flow had been calculated from the temperature difference across a sheet whose thermal characteristics are known. Since convection in a vertical air space has great significance for distribution of heat flow, it is not possible to calculate the flow through the window with any great degree of accuracy. By laying down the test specimen and creating a heat flow vertically downwards through the specimen, no convection will occur. The test specimen consists of 6 mm plexiglass + air space + 6 mm plexiglass and has overall dimensions of 275×275 mm (see FIG.5.4.1). The spacer profiles are wood or extruded polystyrene (Roofmate). The width of air space was $d=10, 20$ and 50 mm.

Heat flow through the horizontal air space is made up by conduction and radiation. These may be expressed as follows.

$$P_C = \frac{\lambda(T_H - T_C)A}{d} \quad (5.6)$$

$$P_R = \epsilon_{\text{RES}} \sigma A(T_H^4 - T_C^4) \quad (5.7)$$

Let us assume that ϵ_n for plexiglass is 0.92 which, according to Eckert (1972), gives $\epsilon_{\Delta} = 0.92 \cdot 0.95 = 0.88$, and $\epsilon_{RES} = 0.79$.

We then have

$$\frac{P_{cR}}{A} = \frac{\lambda(T_H - T_C)}{d} + 0.79 \cdot 5.67 \cdot 10^{-8} (T_H^4 - T_C^4)$$

The thermal resistance is then

$$\begin{aligned} m = \frac{\Delta T A}{P_{cR}} &= \frac{T_H - T_C}{\frac{\lambda(T_H - T_C)}{d} + 4.49 \cdot 10^{-8} (T_H^4 - T_C^4)} = \\ &= \frac{1}{\frac{\lambda}{d} + 0.18 \left(\frac{T_m}{100}\right)^3} \end{aligned} \quad (5.8)$$

The error due to putting $(T_H^4 + T_C^4)/(T_H - T_C) = 4T_m^3$ is very small, for $T_H - T_C = 20$ K the error is 0.13% and at 10 K 0.03%. On the other hand, the profiles around the edges may affect radiant heat transfer between the cold and warm surfaces. Surface temperatures have been measured at the midpoint of the test specimen. The angular coefficient F_{12} is then

d	A/H=B/H	F_{12}
0.01	12.2	0.988
0.02	6.1	0.972
0.05	2.4	0.876

If the temperature of the spacer is the mean temperature between the surfaces, for air space widths of $d = 0.01, 0.02$ and 0.05 m the error in the term P_R will be -0.6%, -1.4% and -6.0% respectively. This means that the thermal resistance increases by about the same order.

The air in the space may be assumed to have a high relative humidity. The panes were assembled at room temperature and then cooled. If the room is at $+22^\circ\text{C}$ and 25% relative humidity, the air will be saturated at about 0°C . No condensation was noted

between the panes during measurements. However, the space between panes is so little that no appreciable quantity of moisture is precipitated. The volume of the space is a maximum of 3 dm^3 and contains approximately 5 g of moisture per m^3 , i.e. the maximum quantity of moisture in the space is 15 mg . Only a relatively small portion of this will however be precipitated.

The thermal conductivity of dry air at atmospheric pressure (760 mm Hg) and 0°C is 0.0241 W/mK . At other temperatures the thermal conductivity is obtained from the expression

$$\lambda_m = -2.8 \cdot 10^{-4} + 9.92 \cdot 10^{-5} T_m - 3.6 \cdot 10^{-8} T_m^2 \quad (\text{W/mK}) \quad (5.9)$$

where

$$T_m = \text{mean temperature in the air space (K)}$$

The relationship has been modified on the basis of tabulated values according to Eckert (1972) for temperatures between 200 and 400 K . The relationship is

$$\frac{d\lambda_m}{dT_m} = 9.92 \cdot 10^{-5} - 7.2 \cdot 10^{-8} T_m \quad (5.10)$$

and at 0°C $d\lambda_m/dT_m = 7.95 \cdot 10^{-5}$.

Alteration of the mean temperature in the air space has little effect on thermal conductivity. If it is changed from 0°C to -5°C , thermal conductivity decreases by 1.7% .

At room temperature thermal conductivity is affected only very marginally by the moisture content of the air.

The following relationship is used for calculation of the thermal resistance of the horizontal air space

$$m = \frac{1}{\frac{0.0241 + 8 \cdot 10^{-5} T_m}{d} + 0.18 \left(\frac{T_m + 273.15}{100} \right)^3} \quad (5.11)$$

where

t_m = mean temperature in the air space ($^{\circ}\text{C}$)
 d = width of air space

For a mean temperature of 0°C thermal resistance varies with d as follows:

$d(\text{m})$	$q(\text{W}/\text{m}^2)$	$m(\text{m}^2\text{K}/\text{W})$	$\lambda_{\text{ekv}}(=d/m)$
at $\Delta t = 10^{\circ}\text{C}$			
0.01	60.6	0.165	0.061
0.02	48.8	0.205	0.097
0.03	44.6	0.224	0.134
0.05	41.5	0.241	0.208
0.10	39.1	0.256	0.391
	36.8	0.272	-

Heat flow is determined by measuring the temperature difference across a plate and multiplying this by a calibration constant. This constant can be obtained by dividing the thermal conductivity of the plate by its thickness. One of the difficulties is correct measurement of the plate thickness or, to be more precise, the thickness over which the surface sensors are mounted. The thermal conductivity has been measured by Gullfiber AB, the value $\lambda = 0.195 \text{ W/mK}$ having been obtained for the plexiglass at a mean temperature of $+10^{\circ}\text{C}$. It is assumed that thermal conductivity changes by about -0.1% per $^{\circ}\text{C}$ as the mean temperature alters. The temperature between the sheets of plexiglass is $18\text{--}30^{\circ}\text{C}$ depending on the convective coefficient of heat transfer at the surfaces (e.g. a fan). This means that thermal resistance between the sheets of plexiglass changes by $0.003\text{--}0.004 \text{ m}^2\text{K}/\text{W}$ at $d=0.01 \text{ m}$, and by $0.004\text{--}0.006 \text{ m}^2\text{K}/\text{W}$ at $d=0.05 \text{ m}$. This temperature dependence has been corrected for in TAB.5.4.1.

When the values of the plate thickness (d) and thermal conductivity (λ) are known, density of heat flow rate (q) can be determined from

$q = \Delta t \cdot \lambda / d$, where Δt = temperature difference across the plate.

Since the temperature difference across the plate is small ($2-4^{\circ}\text{C}$), it is of the utmost importance that the sensors measure a correct surface temperature. Great care has therefore been taken in attaching the sensors to the sheets of plexiglass. These had been glued with epoxy adhesive (Araldite) and then covered by woven tape, since the sensors must be in perfect contact with the surface and must not be displaced from their position. The adhesive layer must be thin and the sensors must also be thin and have a large contact surface. The surface sensors are copper-constantan thermocouples, with the wire 0.1 mm thick and 1.5 mm wide. The datalogger measures temperatures in tenths of degrees, but a mean value of improved accuracy is obtained by making measurements over an extended period. In order that the sensors on each side of the plexiglass may be compared, these were immersed in a water bath. The temperature of the water bath was varied between 5°C and 20°C . During measurements the bath was kept in vigorous circulation to ensure a uniform temperature. The temperature difference across the plate so obtained for the same ambient temperature was used to correct the heat flows.

The test specimen was mounted on a freeze box instead of the lid. Conditions on the cold and warm surfaces were varied by altering the air flow rate. The convective heat transfer was either in the form of natural convection or forced convection created by a fan which had an air flow rate of approximately 1 m/s .

It is seen from TAB.5.4.1 that for $d=50\text{ mm}$ the thermal resistance measured at the upper and lower sheet of plexiglass was approximately the same irrespective of the spacer material and the presence of fan on either the warm or cold side. The mean value of m_{UPPER} and m_{LOWER} is constant and about 16-19% below the calculated thermal resistance (m_{CAL}). For $d=10$ and 20 mm and a wood spacer the difference between m_{UPPER} and m_{LOWER} is greater. The mean value of these is approximately 16-18% less than the calculated value (compare with $d=50\text{ mm}$). For a spacer of an insulation material and a width of $d=10\text{ mm}$, the mean of m_{UPPER} and m_{LOWER} is only about 6% less than the calculated value. It is difficult to explain why the difference is so slight in this case, but if it is not due to an error in measurements it may be caused by the creation of temperature differences

near the spacer with subsequent air movements in the air space.

5.4.2 Heat flow sensors

Heat flow can be measured by attaching heat flow sensors of the auxiliary wall principle to different calibration elements. In actual fact it is a temperature difference across the heat flow sensor which is measured by a thermopile, and since material data relating to thermal conductivity (λ) are known, the heat flow can be calculated. It is in addition assumed that flow is unidimensional when the temperature difference between the outer and inner surfaces is measured at the flow sensor. This assumes that heat flow from the warm to the cold surface takes place perpendicular to the surface. In this way no account is taken of the fact that a large proportion of the heat flow occurs by convection which creates a rotary air movement inside the air space. It is not known whether the disturbing influence of the heat flow sensor on surface temperatures and heat flows is insignificant or of critical importance. The thermal conductivity (λ) of the underlying material is of great significance, if λ increases, the measured heat flow decreases. Some of the error due to this can be eliminated by increasing the protective zone of the meter. This reduces or eliminates the deformation of the heat field around the edges of the meter. Other sources of error which may arise in conjunction with measurements with a heat flow sensor are as follows: an air space may be created between the surface and the sensor when this is taped on, and it is not known what scatter there is between readings given by different sensors, whether the results obtained during different measurements will be the same, whether the measures value agrees with that calculated theoretically, etc.

When a heat flow sensor is attached to a surface, it increases the resistance of the surface by

$$m = \frac{0.003}{0.20} = 0.015 \text{ m}^2\text{K/W}$$

For a window construction the total resistance increases by 4%. For a badly insulated construction this is a relatively large change. In addition there is a risk that the heat flow sensor will disturb heat transfer through the window and convection on the inside of the window. Experiments have been made with flow sensors attached to different types of construction with different types of surface cladding and insulation. The results were contradictory, which created great difficulties during evaluation. Heat flow sensors have also been studied in conjunction with measurements of the temperature difference (Subsection 5.4.1) on a horizontal test specimen. The thermal resistance measured with the flow sensor was 11-16% larger than that calculated (TAB.5.4.1).

It is seen in Subsection 6.7.1 that the thermal resistance measured by the flow sensor was 11-23% larger than the calculated value. The construction consisted of 4 mm glass, 50 mm mineral wool insulation and 4 mm glass, the insulation being calibrated in a Lang apparatus. The causes of this difference may be as follows:

1. The heat flow measured by the heat flow sensor is wrong
 - a) the calibration coefficient given by the maker may have changed
 - b) the boundary conditions of the sensor are different from those during the original calibration
2. The actual flow is greater than the calculated flow
 - a) the value of λ is wrong
 - b) the actual flow in the construction is different from that calculated

For 1a, 2a and 2b the correction necessary is independent of the magnitude of heat flow. In case 1b the sensors are calibrated between solid materials, while measurements are carried out with one side in contact with air. An increased flow through the construction gives rise to a change in convection and the radiant heat transfer with the surroundings. It is seen from TAB.5.4.1

that the difference between calculated and measured thermal resistance for a horizontal air space increases from 11% to 16% when a fan is directed towards the sensor.

It can also be seen from Subsection 6.7.1 that the corrected thermal resistance for the air space ($m_{\text{space}} = 0.175 \text{ m}^2\text{K/W}$) is of the same order as the value measured in the hot box ($m_{\text{space}} = \text{approx } 0.18 \text{ m}^2\text{K/W}$). This indicates that the correction gives approximately the correct value of the flow.

TAB.5.1.1 Calibration tests for 1.2x1.2 m and 0.918x1.2 m window openings and for test windows P1, P2, P3, P4, P5, P6 and P7. T=wood fibre board, I=insulation (Gullfiber 1275 facade sheet), G=glass. Assumed variation in thermal conductivity is 0.0020 W/K°C for test windows P1-P7, 0.0015 W/K°C for 100 mm insulation, and 0.0005 W/K°C for the frame. The values for fabric loss through the frame and for window P5 are calculated on the basis of the measurements.

	$\dot{q}_m$	Fabric loss (W/K) through wall+frame			Fabric loss through frame (W/K)
		measured	corr.to $\dot{q}_m=0$ °C	mean	
3T+100I+3T 1200x1200	8.4 -1.7 -1.8	0.475 0.376 0.463	0.462 0.379 0.466	0.436	
3T+100I+3T 918x1200	10.4 5.3 -1.7	0.532 0.554 0.493	0.516 0.546 0.496	0.517	
4G+50I+4G 1152x1150 P1	10.3 5.2 -2.4	0.654 0.648 0.635	0.633 0.637 0.640	0.637	0.637-0.436=0.201 (k=1.75 W/m ² K)
3G+50I+3G 866x1150 P2	10.5 5.4 -2.4	0.714 0.709 0.678	0.693 0.698 0.683	0.691	0.691-0.517=0.174 (k=1.65 W/m ² K)
4G+50I+4G 585x1150 P3	10.4 5.2 -2.3	0.724 0.703 0.706	0.703 0.692 0.711	0.702	assume k=1.7 W/m ² K ⇒ 0.518 W/K
4G+50I+4G 300x1150 P4	10.4 5.1 -2.3	0.714 0.749 0.719	0.693 0.738 0.724	0.718	assume k=1.7 W/m ² K ⇒ 0.125 W/K
4G+50I+4G 855x1137 P6 (P7)	5.3 -2.2	0.717 0.697	0.706 0.702	0.704	0.704-0.517=0.187 (k=1.44 W/m ² K)
3T+50I+3T 855x1137 P6 (P7)	10.4 5.3 -2.3	0.776 0.688 0.676	0.755 0.677 0.681	0.709	
P5	-	-	-	0.517+0.055 =0.572	assume k=0.52 W/Km ² → 0.055 W/K

TAB.5.3.1 Calibration tests for different methods of attachment to a glass surface. The deviation has been calculated as $\Delta\dot{v}_m/\Delta\dot{v}_{TOT}$, where $\Delta\dot{v}_m = \dot{v}_{meas} - \dot{v}_{CAL}$ and $\Delta\dot{v}_{TOT} = \dot{v}_{air} - \dot{v}_{water}$. $\Delta\dot{v}_{TOT}$ has been varied between $-4.5\text{ }^{\circ}\text{C}$ and $20.1\text{ }^{\circ}\text{C}$. For temperature differences of 20 to $4\text{ }^{\circ}\text{C}$, the calculated values of m_i are 0.114-0.133 $\text{m}^2\text{K/W}$. L (mm) denotes the distance over which the sensor is in contact with the surface.

	L (mm)	Deviation (%)	No
<u>Thermocouple wire (copper constantan)</u>			
Masking tape	25	11.8	1
	45	12.7	8
	45	16.0	8
	70	11.5	1
Woven tape	45	10.3	1
Woven tape, spiral	200	10.5	1
Double sided tape	50	12.0	2
Woven tape, peeled, spiral	130	16.4	7
Masking tape, peeled	60	11.1	7
Masking tape, silicon	50	3.4	5
Aluminium tape	55	0.4	8
Plastic Padding	15	9.4	1
Plastic Padding	45	3.7	8
Araldite	45	6.2	1
Araldite, peeled, spiral	130	3.0	8
School glue, peeled, spiral	130	3.8	8
Contact adhesive, peeled	60	4.1	7
<u>Surface sensors</u>			
Masking tape	45	4.7	9
Woven tape	45	3.9	7
Double sided tape	20	2.2	6
Masking tape, silicon	45	1.8	12

TAB.5.4.1 Comparison of measured and theoretically calculated values of the thermal resistance of the air space.

m_{UPPER} , m_{LOWER} , m_{MEAN} =calculated thermal resistance of air space at the top and bottom sheet of plexiglass, and the mean value of these. Fans above (a) or below (b) the test specimen.

m_{CAL} =theoretically calculated thermal resistance of air space according to Equation (5.11).

m_{FL} =thermal resistance measured with heat flow meter.

No	Space width (mm)	λ of spacer	Fan	Thermal resistance of air space (m^2K/W)					diff (%) (MEAN-CAL)
				m_{FL}	m_{UPPER}	m_{LOWER}	m_{MEAN}	m_{CAL}	
2	10	0.04	a+b		0.146	0.160	0.153	0.162	-5.7
1	10	0.14	a+b		0.127	0.144	0.136	0.166	-18.1
1	10	0.04	-		0.150	0.156	0.153	0.164	-5.5
1	20	0.14	a+b		0.158	0.177	0.168	0.202	-16.8
1	50	0.04	a+b		0.189	0.200	0.195	0.236	-17.4
14	50	0.14	a+b	0.274	0.193	0.198	0.196	0.237	-17.3
8	50	0.14	b	0.282	0.208	0.205	0.207	0.255	-18.8
1	50	0.14	a		0.179	0.184	0.182	0.220	-17.3
7	50	0.14	-		0.192	0.198	0.195	0.239	-18.4

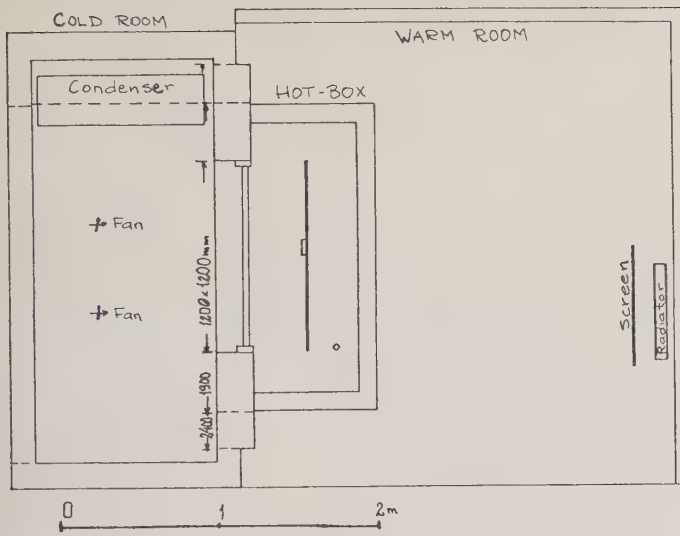


FIG.5.1.1

Section through the cold room, measuring box and warm room.

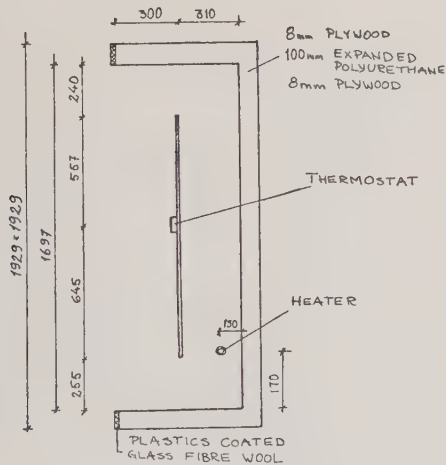


FIG.5.1.2

The measuring box.

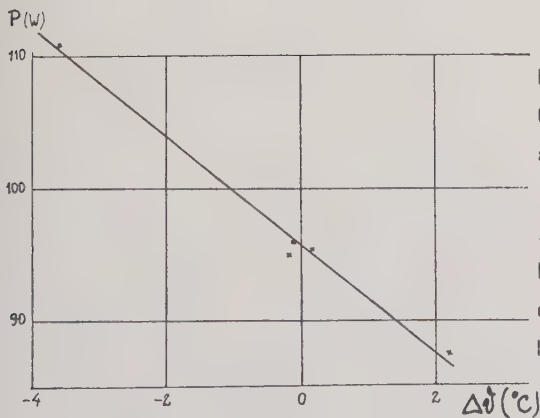


FIG.5.1.3

Correlation between the power P (W) and the temperature difference ΔT ($^{\circ}\text{C}$) across the wall of the measuring box. The regression line is $P = 95.74 - 4.122\Delta T$ $R = -0.996$. Calibration element K2. The temperature difference between the cold and warm room was 45°C .

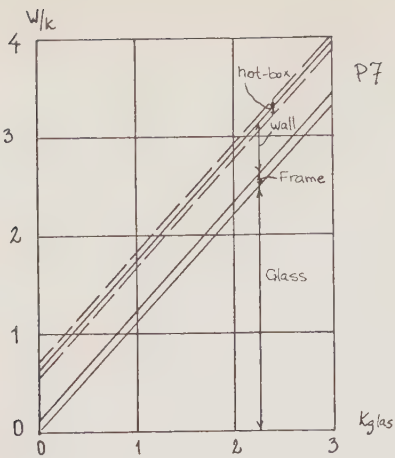


FIG.5.1.4

Order of magnitude of flows through glazed portion, frame, wall and box wall for varying k -values for the glazed portion in test windows P7 (outside dimensions 918x1200 mm).

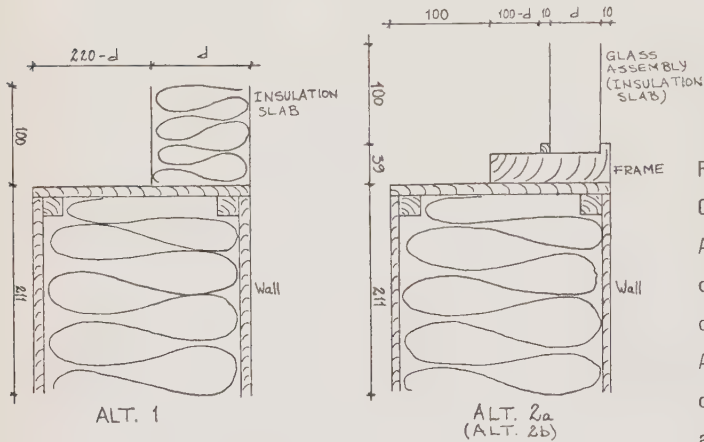


FIG.5.2.1

Calibration calculations.
Alt.1: insulation slabs set directly into the window opening (1200x1200 mm).
Alt.2: frame (39x120 mm or 39x200 mm) with glass assembly (alt 2a) or insulation slab (alt 2b).

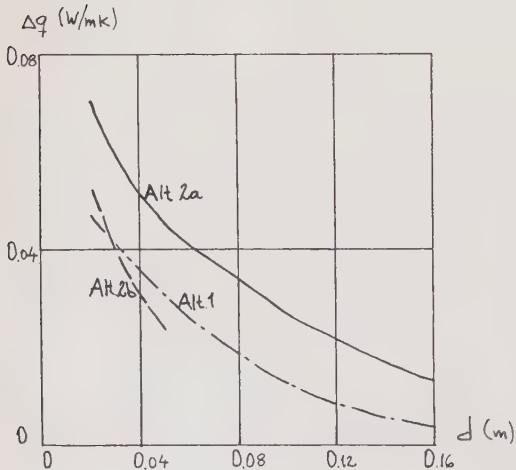


FIG.5.2.2

Comparison of calculated flow and uni-dimensional flow per metre of wall.

$q/l = (k_{CAL} - k_{DIM})$. d = thickness of insulation slab (Alt 1, 2b) or width of air space (Alt 2a).

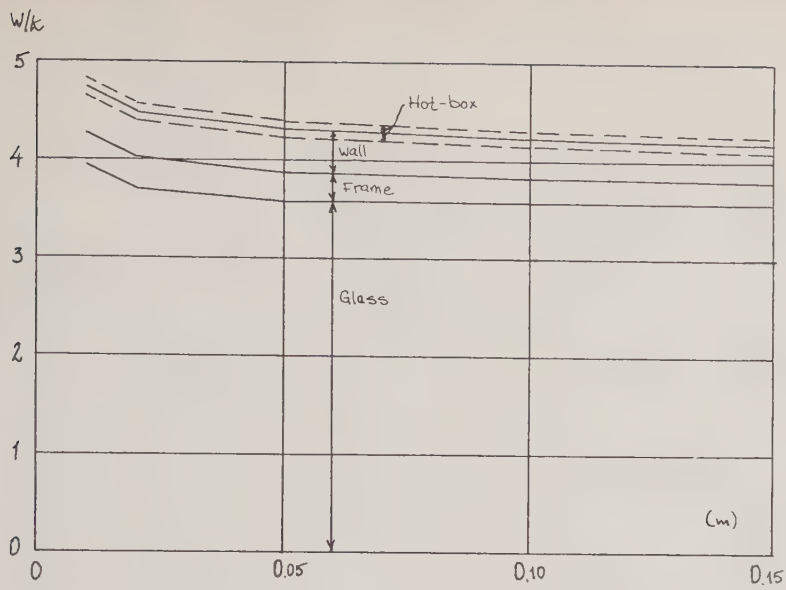


FIG.5.2.3 Influence of air space width on power supplied to the hot box. Window construction according to Alt 2a in FIG.5.2.1 with 1200x1200 mm window opening.

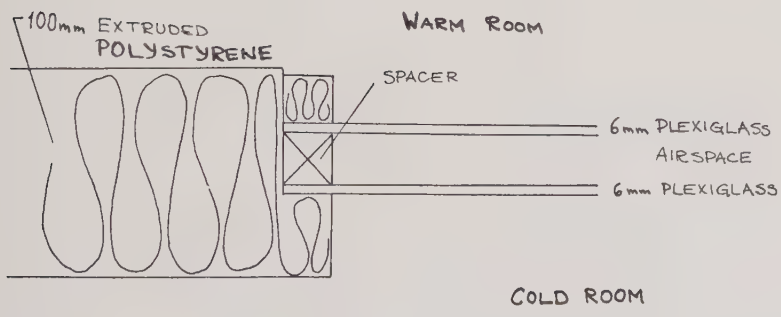


FIG.5.4.1 Test arrangement for plexiglass unit with spacer of wood or extruded polystyrene.

6 EXPERIMENTAL MEASUREMENTS

6.1 GENERAL

6.1.1 Heat flow through a vertical space

Climatic conditions can be simulated by creating different thermal conditions along the surfaces of a window. For a vertical air space, the Rayleigh number (which indicates the degree of convection), the height/width ratio and the Prandtl number (which is determined by the physical properties of a gas) can be varied. In addition, the conditions along both the vertical and horizontal edges can be varied. For the vertical surfaces, surface heat transfer is a function of both convection, i.e. the type of flow (laminar or turbulent) and of radiation conditions i.e. the emissivities of glass surfaces and surrounding surfaces. The temperature around the vertical and horizontal edges of the air space depends on the materials used and the profile construction.

Some special windows have been designed in order that the effect of these factors may be studied (FIG.6.1.1-6.1.3). The designations for the spaces and panes of the window are set out in FIG.6.1.4.

The thermal transmittance (k) at a point on a glass portion is calculated by aggregating the thermal resistance of the individual components, e.g. for a double glazed window

$$\frac{1}{k} = m_u + m_{gl} + m_{sp1} + m_{gl} + m_i$$

where

m_u, m_i = surface resistance on the outside and inside respectively

m_{gl} = resistance of glass

m_{sp1} = resistance of space

The surface resistance comprises a convective/conductive and a radiant component, where

$$m_i = \frac{1}{\alpha_{ic} + \alpha_{iR}} \qquad m_u = \frac{1}{\alpha_{uc} + \alpha_{uR}}$$

The radiant component is a function of the emissivities of the glass surface, the room and sky+ground, the temperatures of the glass surface, room surfaces and the surfaces outdoors. The convective component is a function of the character of air movement (laminar or turbulent) and varies with the velocity and direction of air movement and with the temperature difference between the glass surface and the air. This component may thus vary a lot depending on climatic conditions on the outside or air movements inside due to the ventilation system, heating system and the geometrical configuration near the window. For a single glazed window for which the thermal transmittance k is very nearly identical to the surface resistance the k -value may fluctuate by up to $\pm 40\%$ about the mean value depending on variations in m_i and m_u .

Measurement of heat flow for air spaces of variable width and orientation has been carried out by Robinson, Powlitch (1954). Linke (1956) has made measurements on hermetically sealed glass assemblies. Christensen, Brown, Wilson (1964) have carried out tests on an unventilated window with an air space which was varied between 6 mm and 150 mm.

6.1.2 Measurement of heat flows in test windows P1-P6

As pointed out before in Section 5.4, measurement of heat flows in a window is difficult. According to Subsection 5.4.1, the method of measuring the temperature difference across a pane and calculating the heat flow from this appears to have a consistent error of about 15%. If this error is relatively constant, this means that the mean level of heat flow along a surface will be too high. However, the general shape of the curve is comparatively unchanged. On the whole, heat flows have been determined by measurement of temperature differences and used in studying the way in which the heat flow varies along a vertical line on a glass surface.

For test window P6, measurements according to Section 6.3 have been supplemented by a comparison of different sensors. The test window has been modified for this purpose by rotating the line with the sensors through 90° at the time of each measurement (i.e. four different positions). The heat flow has thus been determined in both the horizontal and vertical directions (with the window turned upside down). The window has also been turned around so that the inside has been turned towards the cold side. It has been found in this way that the sensors have given markedly different values at two points. In the figures these values have been put inside brackets and a correction has been made on the basis of the measurements. The other readings are relatively well grouped. For the inner pane the heat flows are situated within $\pm 3\%$, and for the outer pane within $\pm 5\%$.

6.2 TEST WINDOWS P1-P5

In order to study the way in which the distribution of temperature and heat flow changes when the width of air space and window format is varied, different types of special window have been constructed. These test windows comprise a 4 mm pane of glass on the cold side and a 6 mm sheet of plexiglass on the warm side, both set into a wooden frame. The spacer between the panes is also wood, and the shapes of the profiles are illustrated in FIG. 6.1.1-6.1.2. The width of air space between the panes has been varied between 10, 20, 30, 40 and 50 mm. The width of the test window has been constant at 1200 mm and the width of the air space at 1124 mm. The height of the test windows has been varied between 1200, 918, 638 and 349 mm, with the height of the air space at 1125, 841, 561 and 272 mm. The coordinates x_1/L , y_1/L etc for the air space are set out in Subsection 2.3.3.2.

In test window P5 the wooden frame has been changed to one made of expanded polyurethane. The calibration of the hot box, wall element and test window frame has been described in Section 5.1.

Along vertical lines, the temperature is measured on the cold surface, in the middle of the air space and on both sides of the sheet of plexiglass. The air temperature is measured in the middle of the air space ($z_1/b=0.50$). For practical reasons, the surface temperatures are measured some distance away from the centre line ($z_1/b=0.377$). In this way the vertical temperature profiles are obtained for the air space and its boundaries. Knowledge of the temperature on both sides of the plexiglass, together with material data, enables calculation of heat flow at a point to be carried out.

In addition, temperature sensors have been positioned at the centre of the window ($x_1/L=0.50$, $z_1/b=0.50$) at different distances into the air space ($y_1/d=0.25$, 0.5 and 0.75). The temperature gradient in the horizontal direction can be studied in this way to characterize convection. The vertical temperature profiles change as the flow changes and a transition occurs from one convection regime to another.

The horizontal temperature distribution is illustrated in FIG. 6.2.1-6.2.5. Along the cold and warm sides the gradient is steep, and in the middle of the space the nondimensional temperature is usually $0.5-0.6$. The way the horizontal temperature gradient changes from -0.75 to zero, i.e. a transition occurs from the conduction regime to the boundary layer regime, is shown in FIG.6.2.4. The nondimensional temperature θ is defined as

$$\theta = \frac{\mathcal{T} - \mathcal{T}_{sp1-}}{\mathcal{T}_{sp1+} - \mathcal{T}_{sp1-}}$$

where

$\mathcal{T}$ = temperature in the air space
 $\mathcal{T}_{sp1+}$, $\mathcal{T}_{sp1-}$ = surface temperatures on the warm and cold boundary surface respectively of the air space, as shown in FIG.6.1.4.

FIG.6.2.5 shows the horizontal gradient for test windows P1-P4 in which the two air spaces were each 20 mm. The air space on the warm side has a gradient in all test windows, which indicates very slight convection.

The vertical temperature profiles for window P2 ($L=841$ mm), for different space widths d , are set out in FIG.6.2.6-6.2.10. As the width of space is increased, the constant air temperature in the central portions of the space is increasingly affected by the end regions. An S-shaped curve is first formed and then a more or less straight line, with the temperature varying with distance from the edge. Finally, the vertical temperature profiles for test window P2 with a double air space are shown in FIG.6.2.11. The way the middle pane is affected by both convection currents, downwards on the warm side and upwards on the cold side, can be seen in the figure. In Jonsson (1985), there are diagrams for single and double air spaces in windows with $L=1125$, 561 and 272 mm.

The heat transfer for different air space widths and temperature differences can be illustrated by determining the thermal resistance of the air space or the specific heat flow. The heat flow at a point divided by the air temperature difference gives the value of the specific flow, i.e. a local value of the thermal transmittance (k), which can be compared with the average k -value for the glass surface as a whole. The thermal resistance illustrates conditions in the air space itself. The thermal resistance is calculated as the difference between surface temperatures divided by the heat flow at the same level. Use of the temperatures at the same level provides an analogy with unidimensional heat transfer by conduction. Although unidimensional flow changes as convection increases, in a window it is generally radiation which dominates heat flow.

Once the temperature difference across the sheet of plexiglass is known, the heat flow can be calculated. This flow contains the contributions due to convection, conduction and radiation, i.e. the total flow.

FIG.6.2.12 sets out the local k -value which has been defined as the heat flow through the inner sheet of plexiglass divided by the mean temperature difference between the indoor and outdoor air. These curves are intended for the illustration of the magnitude and distribution of heat losses. It is to be noted that the temperature difference includes the external and internal surface

resistances.

It is seen from the figure that heat flow is greatest in the bottom corner and then decreases with increasing height. In general, heat flow is least in the top corner. The values for $x_1/L = 0.73$ are probably too low (they have no direct counterpart in test window P6) and have been excluded in the figure. As a general trend, the lower part of the curve may be said to be comparatively similar for air space widths of 20–50 mm, while the upper part has a slight tendency to drop as the width increases. For a 10 mm air space the shape of the curve is different, the upper part is practically constant while the bottom part has a lower heat flow than other air space widths.

In order that the contribution made by convection to the total resistance may be studied, the calculated share due to radiation has been deducted in FIG.6.2.13. Between two parallel and infinite sheets, the heat flow due to radiation is

$$P_R = \epsilon_{RES} \sigma A (T_H^4 - T_C^4)$$

where

T_H, T_C = temperature (K) for the warm and cold sides
respectively

$$\sigma = 5.67 \cdot 10^{-8} \text{ W/m}^2\text{K}^4$$

$$\epsilon_{RES} = 0.74$$

If the temperatures of the cold and warm sides do not vary by very much, the radiant heat exchange will not vary on the inner sheet. It is then possible in calculations to use only the opposite temperature sensors. The radiant heat exchange between a point on the warm side (H) and the surrounding cold surfaces (sheet of plexiglass C, upper (U) and lower (L) spacer) is then

$$P_R = \epsilon_{RES} \sigma A (T_H^4 - F_{HC} T_C^4 - F_{HU} T_U^4 - F_{HL} T_L^4)$$

where

T_U, T_L = temperature (K) of the upper and lower boundary

F_{HC}, F_{HU}, F_{HL} = angular coefficient for surface C, U and L respectively as seen from the point H

If there is a large change in the temperature on the cold side, the opposite temperature sensor will not represent the surface temperature but a correction must be made on the basis of adjacent temperatures.

When the share of the heat flow due to radiation has been deducted, the Nusselt number can be calculated for the share due to convection.

$$Nu = \frac{P_T - P_R}{P_{cond}} = \frac{P_T - P_R}{\frac{\lambda}{d}(T_H - T_C)A} = \frac{1/m - P_R/(T_H - T_C)A}{\lambda/d}$$

where

P_T, P_R och P_{cond} = total heat flow and flow due to radiation and conduction respectively

λ, d, A = thermal conductivity (W/mK), width (m) and area (m²) of air space

m = measured thermal resistance (m²K/W)

It is seen from the figure that the shapes of the curves are similar, the only difference being that they are displaced laterally. Certain details are more pronounced, such as the large change at the bottom corner which increases as the air space becomes wider. When the width of air space is 10 mm, there is little convection, and the Nusselt number is almost equal to one.

APP.6.2.1-6.2.5 set out all the cases with test windows P1, P2, P3, P4 and P5. The measured power consumption in the hot box, corrected with respect to fabric losses through the hot box and wall element, can be converted into a value of the air space thermal resistance. See FIG.6.2.14-6.2.17. The influence due to

an increase in air space width is moderate, for P1 and P2 the thermal resistance is almost constant while for P3 and P4 there is a slight rise in thermal resistance as the width increases. For a width of 10 mm the thermal resistance is practically the same for all four test windows. For some formats the curve flattens out at $d=40-50$ mm. The influence due to the height of the air space is very slight. However, there is a general rise in thermal resistance for P3 and P4, i.e. the resistance increases as the height/width ratio (L/d) decreases. This is contradicted by other calculations (e.g. FIG.3.7.12) which state that the Nusselt number increases as L/d decreases. The probable reason for this is that the effect of the boundary conditions is less in the smaller formats (P3, P4) and that the corrections which are made do not fully compensate for this. In FIG.6.2.14-6.2.17 an endeavour has been made in the calculations to remove the effect due to the wooden spacer. Corrections have been made by modifying the thermal transmittance (k), the thermal resistance for the air space and the spacer being put

$$m_{sp}+0.235 \quad \text{and} \quad d/0.14+0.235+0.012/0.14 \quad \text{respectively}$$

This correction assumed adiabatic conditions between the air space and the spacer, i.e. no heat transfer. This is not quite correct, but the correction gives an idea of the influence which the spacer exerts.

In test window P5 which has a frame of expanded polyurethane, the temperatures and flows in the air space are almost the same as in windows which have a wooden frame. For practical reasons it was not possible to calibrate test window P5 according to Section 5.1. This value was instead calculated with reference to test window P2 in which the loss through the frame is 0.174 W/K . This is equivalent to a thermal transmittance (k) of $1.65 \text{ W/m}^2\text{K}$. Under unidimensional heat flow conditions this corresponds to a wood thickness of 0.05 m . Using this thickness and the thermal conductivity for expanded polyurethane ($\lambda=0.030 \text{ W/mK}$), the thermal transmittance (k) obtained is $0.52 \text{ W/m}^2\text{K}$, i.e. a fabric loss of 0.055 W/K .

In FIG.6.2.18 test windows P1-P4 have been classified into the

three convection regimes. On comparing this with the investigations by Yin, Wung, Chen (1978), the agreement is comparatively good.

6.3 TEST WINDOW P6

Test window P6 was constructed in order that the temperature distribution and the heat flows on the warm and cold sides may be studied. This window consists of two 6 mm sheets of plexiglass, one on the warm side and one on the cold side, set into a wooden frame as shown in FIG.6.1.2. The overall size is 918x1200 mm, and the size of air space is 840x1122 mm. The spacer between the sheets of plexiglass is wood, its width being changed to give air space widths of 10, 20, 30, 40, 80 and 150 mm. For the last of these widths an additional wooden frame was fitted on the cold side. Comparisons between measurements with and without this additional frame show that the effect due to the frame is negligible, in any case the measuring accuracy is not enough to detect any differences.

Along the vertical centre line, temperatures are measured on both sides of the sheets of plexiglass. In this way the heat flows can be calculated. The air temperature in the space was measured at distances of $d/4$, $d/2$ and $3d/4$ from the warm side of the air space, where d is width of air space. The sensors are placed at midheight but have a slight lateral displacement (65 mm) in relation to the centre line. In these tests few temperature sensors have been used inside the air space in order that convection in the air space should not be disturbed too much. The thermal conductivity of plexiglass changes with temperature, and the calibration of the inner and outer sheet should in actual fact be different for the same thickness. Since however this temperature dependence is comparatively slight and the main purpose is to compare the characteristic features of temperatures and heat flows, no correction has been made with respect to this temperature dependence.

6.3.1 One air space

FIG.6.3.1-6.3.2 set out the vertical temperature distribution for different air space widths. The shape of the curves, the s-shaped profile for the inner sheet and the semi-s shape for the outer sheet, agrees with other measurements. The temperature has been converted into the nondimensional parameter θ , where $\theta=1$ for indoor air and $\theta=0$ for outdoor air.

The specific flow shown in FIG.6.3.7-6.3.9, which may be said to relate to the local thermal transmittance (k) at the point, has a relatively uniform profile on the inside which decreases towards the upper edge. At the outer sheet the local k -value increases with distance from the bottom corner. The flow at the upper corner is almost twice as large as at the bottom corner. If the mean values of the thermal resistance, calculated across the inner and outer sheets of plexiglass (APP.6.3.1), are compared it is seen that the difference between these increases as the width of air space becomes larger. This indicates that the flow through the inner sheet of plexiglass is greater than through the outer one. The explanation for this may be that surface heat transfer between the air in the space and the frame increases as the width of space and thus the exposed frame surface increases.

The width of the air space has generally been uniform, but measurements have been made on an air space which varies linearly in the vertical direction from 10 to 30 mm. These are shown in Jonsson (1985). The mean values calculated for the air space show small differences. Air spaces of variable width have a slightly lower thermal resistance.

The surface coefficients of heat transfer along the inner and outer surface have great significance for the distribution of temperatures and heat flows. Measurements have therefore been made for air space width $d=40$ mm with no fans at all, fans on the cold side and fans on both the warm and cold side. FIG.6.3.8 shows the effect due to the fans. The heat flow increases when the fans on the warm and/or cold side are turned on. The main reason for this is that the surface resistance decreases. In addition, the resistance of the air space is slightly reduced

when conditions along the edges become more turbulent.

Mean values of temperatures and heat flows are given in APP.6.3.1-6.3.4. The tables show the way in which resistance changes when the mean temperature across the air space changes. Very approximately, a 5 °C change in mean temperature gives rise to a 5% change in thermal resistance.

The effect of air space width on thermal resistance is summarized in FIG.6.3.12. A correction has also been made for the change in fabric loss through frame and wall according to Section 5.2.

6.3.2 Two or more air spaces

When there are two air spaces, the construction, taken from the inside, consists of 6 mm plexiglass, air space, 6 mm plexiglass, air space and 4 mm glass. The spacer is still made of wood. It is evident from FIG.6.3.3-6.3.4 that the vertical temperature distribution on the inner and outer sheet is about the same as for one air space. The temperature distribution on the intermediate sheet is s-shaped, and the temperature difference between the upper and lower edge is greater than at the inner sheet. The specific flow is set out in FIG.6.3.9. The tendency is for the flow along the inner sheet of plexiglass to decrease as the distance from the bottom rail increases. The sheet between the air spaces has an almost constant specific heat flow of approximately 1.6-1.7 W/m²K. In FIG.6.3.13 the thermal resistances of two air spaces have been summarized. If the width of the outer air space is changed, this has only a slight effect on the inner thermal resistance.

For three air spaces, the construction consists, taken from the inside, of 6 mm plexiglass, air space, 6 mm plexiglass, air space, 6 mm plexiglass, air space, and 4 mm glass, the width of air space being either 10 or 20 mm. These measurements are set out in FIG.6.3.5 and 6.3.10. The thermal resistance of the warm air space is due mainly to the width of this air space. If the widths of the other air spaces are changed, this has only a marginal effect on the thermal resistance of the warm air space. The

heat flow along the inside of the window decreases slightly as the distance from the bottom corner increases. The heat flow between two air spaces is almost constant along a vertical line (FIG.6.3.9).

Four air spaces consist of three sheets of plexiglass and two panes of glass, with intermediate air spaces of 10 and 20 mm width (FIG.6.3.6, 6.3.11). The characteristics of these constructions are the same as those with fewer air spaces.

6.3.3 Two and three dimensional flow

Eckert, Drake (1959) state that flow in natural convection is laminar if $Gr_x Pr < 10^8$, and that the heat transfer is then

$$Nu_x = 0.411 (Gr_x Pr)^{1/4} \quad \text{or}$$

$$\alpha_x = \frac{\lambda}{l} 0.411 (Gr_x Pr)^{1/4}$$

When $Gr_x Pr > 10^9$, flow becomes turbulent and heat flow can be described by the following expression

$$Nu_x = 0.10 (Gr_x Pr)^{1/3} \quad \text{or}$$

$$\alpha_x = \frac{\lambda}{l} 0.1 (Gr_x Pr)^{1/3}$$

Between these two values there is a transition zone for $10^8 < Gr_x Pr < 10^9$. At the leading edge of the plate there is extensive convective heat transfer but this decreases all the time within the laminar region (see FIG.6.3.14). In the turbulent region surface transfer of heat is constant and slightly larger than the minimum value in the laminar region.

Since most calculations assume that convection is two dimensional, it may be of interest to find whether this assumption is true. Mallinson, de Vahl Davis (1972) have employed a numerical solution method for the determination of convection in a space of height=1, length=5, depth=1, $Pr=1$, and $Ra=10000$. By studying the traces of particles released near the side walls, it can be seen

that three dimensional flow exists over the entire space. The calculations showed that the thermal effects of the three dimensional flow are confined to regions which are less than one space thickness distant from the side edges.

By means of a laser doppler anemometer, Morrison, Tran (1978) measured velocities in a three dimensional space. The vertical velocity is similar within $0.2 < z < 0.8$ for uninsulated side surfaces, and within $0.1 < z < 0.9$ for insulated ones. A three dimensional movement is also generated due to losses through the side wall (see FIG.6.3.15). Flow appears to move in a spiral from the side surface towards the centre. When the two converging flows have met at the centre, the spiral widens and the two outer spirals outside the central part move towards the side surfaces. Comparisons with other results show that the degree of insulation along the horizontal edge has only a slight effect on the vertical flow.

6.4 MEASUREMENTS ON HERMETICALLY SEALED UNITS CONTAINING GASES AND LOW EMISSIVITY LAYERS

6.4.1 Change of thermal resistance

Since about two thirds of the total heat transfer between two panes of glass consists of radiation, this presents a great potential for the improvement of the thermal insulation of a window. Transparent layers of high transmittance for solar radiation and low emissivity for long wave radiation (selective layers) can be applied to glass or plastics. Plastics films which are used for solar control have low emissivity (reduce the thermal transmittance) and transmit little light. Coatings may consist of gold, silver, copper, aluminium, indium oxide or zinc oxide, see e.g. Selkowitz (1979), Ribbing (1982).

For satisfactory utilization of solar heat, transmission of light must be good while emissivity with respect to long wave radiation should be small. In general, transmittance decreases as emissivity decreases. At present there is a limit at $\epsilon = 0.12$ and a transmittance of 70%.

Since conduction and convection interact in the physical heat transfer process, these two factors must be treated together. The convective heat transfer can be expressed by

$$\alpha_c = \frac{\lambda}{d} Nu_d$$

and

$$Nu_d = D(Gr)^A (Pr)^B (L/d)^C$$

It is however impossible to vary the different factors on their own in order to make α_c a minimum. If one factor, e.g. d , is changed this affects L/d , the Grashof number and d in the expression for α_c . A change may also occur to another convection regime which means that the values of the constants A , B , C and D will be different.

On theoretically grounds, there are two methods whereby the proportions due to convection and conduction can be changed, namely a change of medium in the space and a change of the geometrical configuration of the space. A change of medium implies the use of other gases or gas mixtures, but also a change of pressure in the space. Geometrical configuration can be changed by appropriate subdivision of the space so that convection is reduced. One good way is to subdivide the space into a number of vertical spaces connected in series (vertical subdivision).

If a layer which is transparent in the infrared region (for instance certain plastics) is introduced between two panes of glass, the air space is divided into two separate spaces with regard to convection while radiant heat transfer takes place between the panes of glass. In such cases involving combined layers of different properties, it is appropriate to apply an analogy with electrical quantities. We then have

temperature ($^{\circ}\text{C}$) - voltage (V)
 thermal resistance (K/W) - resistance (ohm)
 heat capacity (Wh/K) - capacitance (F)
 heat flow (W) - current (A)

An equivalent representation of the window construction is given by means of an equivalent RC network.

6.4.2 Test window P7

As pointed out before, heat transfer through a window is a very complex process. If the connection between frame and glass is changed, as regards either materials or configuration, this obviously affects primarily the conditions at the connection, but parts further away may also be subjected to secondary effects. If the conditions relating to a certain part are changed, this will, to a greater or lesser extent, affect the mean values for the construction as a whole. Naturally, this series of measurements can be compared with other test windows, but this may be difficult since several changes are involved as a rule. Comparisons within the same series are however easier to carry out and the results are easier to set out.

For this test series, a total of 14 sealed units of the format 855x1137 mm were obtained from Pilkington AB, Halmstad. The panes had been made by Combiglas AB, Vetlanda. For glass assemblies comprising two panes, the distance between panes has been varied. Different gases such as air, argon and sulphur hexafluoride have been used. In addition, assemblies comprising both two and three panes contained low-emissivity layers. The designation D4-12 denotes D=double glazing (T=triple glazing), 4=thickness of glass in mm and 12=distance between panes in mm. The low-emissivity layer consists of a multilayer with copper as the main constituent, designated "Kappaenergi".

Most measurements have been made on sealed units of 855 mm height, but some panes of 1137 mm height have also been included so that differences, if any, may be seen. The sealed unit was set into the frame as shown in FIG.6.1.3. In order that conditions may be uniform, it was decided to clamp the sealed unit between strips of wood. Round the sealed unit there is a gap of 3 mm between the frame and the unit. In this way the strips of wood exactly coincide with the spacer in the sealed unit, i.e. a glazing rebate of 13 mm. The overall dimensions of the air space are

exactly the same as the overall dimensions of the transparent portion. Some measurements have also been made to ascertain the effect due to alteration of this glazing rebate dimension. In order to ensure that the window is absolutely airtight, tape was applied between frame and wall and between frame and sealed unit both on the inside and outside.

As before, surface temperature sensors have been attached to the glass surface with epoxy adhesive. Most of the sensors have been attached along the vertical centre line, and also along vertical lines at distances of 140 mm from each other. In addition, three heat flow sensors have been placed in the vicinity of the vertical centre line. These have been used for comparison of values determined by energy measurement and by the heat flow sensors. The primary interest has been comparison of different trends due to the different methods of measurement, and not the exact values. A 50 mm Gultiber 1275 facade sheet surrounded by panes of glass has been used for calibration of the wooden frame and wall element. The procedure has been described in Section 5.1.

Surface temperatures along the vertical lines are shown in FIG. 6.4.1-6.4.9 for some sealed units. Generally speaking, the difference between vertical lines is small even though some larger deviations may occur. An example for case D4-12,Ar, is shown in FIG.6.4.5.

On the warm side, the shape of the temperature profile is largely the same for all sealed units. In the centre the temperature is approximately constant, and it increases slightly towards the upper part. At the bottom edge there is a drastic drop in temperature, but the minimum temperatures are relatively similar. They vary between 0.5 and 0.6, and there is a tendency for temperatures to rise as the thermal resistance increases. However, a surface temperature of 0.5-0.6 implies that the internal surface resistance constitutes a large proportion of the total resistance. In calculations for unidimensional heat flow, such a value of the surface temperature results in a thermal resistance of 0.05-0.10 m²K/W inside the sealed unit. Along the edges of the glass, the panes are connected by a spacer of aluminium. Since this material is a very good conductor of heat, the resistance of

this part consists almost completely of the surface resistance. It is consequently this spacer which gives rise to the low temperatures that cause condensation problems. Alteration of the frame construction in such a way that the glazing rebate is increased gives rise to only marginal changes, an increase of the rebate by 5 mm causes no change.

On the cold side the temperature is practically constant, it is only in the upper part that the temperature rise. This is due to the fact that upward movement inside the space brings warm gas to the top which then begins to cool down at the top on the cold side.

APP.6.4.1 sets out mean values of surface temperatures, the thermal resistance of the air space, heat transfer and the thermal transmittance (k). Minimum temperatures on the inside are also given. Owing to the fact that attachment of the sensor in the corner between glass and wooden strip had been difficult, the next one above this has also been included. All temperatures have been converted into nondimensional factors according to the expression

$$\theta = \frac{t - t_u}{t_i - t_u}$$

These values can again be converted into $^{\circ}\text{C}$ units using the selected inside and outside temperatures.

FIG.6.4.10 shows the way in which the thermal resistance varies with the width of air space. For air and argon the resistance increases linearly with air space width, while for sulphur hexafluoride (SF_6) the resistance is practically independent of the air space width. The reason for this is that the properties of SF_6 are such that convection is of great significance already for small air space widths (see Section 3.6).

The 14 different types have been plotted in FIG.6.4.11, with different outside temperatures shown by the cross hatched areas. The resistance of an air space may be said to increase by $0.02 \text{ m}^2\text{K/W}$ when the gas is changed to argon, by $0.11 \text{ m}^2\text{K/W}$ due to

the introduction of a low emissivity layer, and by $0.20 \text{ m}^2\text{K/W}$ if the space has both a low emissivity layer and argon.

When the height of the air space is changed from 855 to 1137 mm, there is very little change in thermal resistance and surface temperatures. Turning the sealed unit round so that the low emissivity layer is situated at the cold pane of glass has very slight effect on the properties of the sealed unit.

In order to ascertain what influence boundary conditions have, a test has also been carried out with the fans on the cold side turned off. In this way convection changes from forced to natural convection. The typical s-shaped temperature profile is obtained on the cold side also. Naturally, the external surface resistance increases, in this case from 0.05 to $0.13 \text{ m}^2\text{K/W}$, and this increases the entire temperature level on the cold side. However, the thermal resistance of the air space increases by about $0.01 \text{ m}^2\text{K/W}$. The fact that a change from forced to natural convection along the edges causes the air space resistance to increase slightly has been noted before.

6.5 FRAME TEMPERATURES

In order that an idea may be obtained of the way in which temperature is distributed along the frame, measurements have been made on test window P1 with an air space of 40 mm width. The temperatures have been converted into a nondimensional number

$$\theta = \frac{t - t_u}{t_i - t_u}$$

where

t_i, t_u = inside and outside air temperatures respectively
 t = temperature at the point concerned.

The temperature difference has been approximately 30°C . It is seen from FIG.6.5.1 that the temperature of the frame on the cold

side is very uniform, due mainly to the horizontal air current set up by the fans. On the warm side the vertical temperature differences are large, of approximately the same order as those on the warm side of the glass. Inside the space on the surface of the frame, horizontal temperature distribution is almost linear (FIG.6.5.2). At both departure corners, i.e. the upper corner on the warm side and the bottom corner on the cold side, the change in temperature is a little larger. The discontinuous lines in FIG.6.5.2 are assumed values. When the air temperature in the space and the surface temperature on the stile of the frame are compared in FIG.6.5.1, it is seen that there is good agreement between these in the upper part of the space and at the bottom edge. In the lower part of the space the difference is greater.

6.6 THE EFFECT DUE TO THE AIR SPACE BETWEEN COUPLED CASEMENTS

Measurements have been made to see how the thermal transmittance (k) changes when the width of air space between the panes is increased. In addition to this, the external climate, temperature and wind speed may effect the heat flows and temperatures in varying degrees.

Ventilation between the casements in a coupled wooden window must be maintained to prevent moisture problems inside the window. At the same time, ventilation must not be so large that unnecessarily high heat losses occur. When wind speeds are high, the heat flow will increase and the surface temperature on the inside drop. This would seem to imply that there is a great risk of condensation on the inside of the glass. This is not the case, since high wind speeds and low outside temperatures seldom coincide. In addition, the air change rate in the building often increases when there are strong winds, which causes the relative humidity of the room air to decrease.

There are few investigations concerning the influence of ventilation, and those that are available are to some extent contradictory. Adamson, Höglund (1957) state that when the gap between the windows is sealed in an old double glazed wooden window, the resistance of the air space changes by about 13% (the k -value of

the window by about 6%). Blomsterberg (1982) has carried out measurements on a coupled window comprising a double glazed and a single glazed unit. The effect due to the width of air space is moderate, when the width of the space between casements is increased from 1.5 mm to 4 mm, the k-value of the window is changed by 1-2% at a wind speed of 0.2-1.5 m/s. When the wind speed parallel to the window increases by 1 m/s, the k-value very approximately increases by 4%.

The window selected for this investigation was an inward opening wooden window comprising two coupled casements, the inner one with a hermetically sealed unit. The outside dimensions of the window were 1200x1200 mm, and it was installed in such a way that the inside face of the frame was in the same plane as the inner wall surface. A cross section of the window is shown in FIG. 6.6.1.

The temperature was measured in the middle of all four parts of the frame and casement, both on the inside and outside. The temperature of the glass was measured at five points both on the inside and outside, distributed along the vertical centre line. The temperature was also measured in the rainwater channel of the frame (at four points) and in the air space between the bottom rails of the inner casement and the frame.

All the cases studied are set out in APP.6.6.1. The variables which have been varied are the outside temperature, fans on the cold side, the width of air space between the casements, and the width of the gap between frame and casement on the outside. In addition, the air space between the casements has in certain cases been fitted with a dust strip or has been taped. The wind speed perpendicular to the window has been approx 1, 0.5 and 0 m/s respectively for fan designations II, I and Ø. In the last case there is some movement of air due to the fans of the refrigeration plant. This current of air moves along the window from the top downwards. The designations $d_1=7.0$ mm and $d_2=9.4$ mm denote that the mean width of the space between the casements is 7.0 mm and that the mean gap between frame and outer casement is 9.4 mm. On delivery these dimensions were 2.4 and 2.8 mm respectively. The air space between the casements has been widened by

planing the inner faces of the outer casement. In this way the external faces of the casement and frame remained in the same positions (see FIG.6.6.1).

The influence due to temperature, fans and width of air space has been summarized in TAB.6.6.1. The effect of outside temperature on the thermal transmittance (k) is very small. An increase in wind speed slightly increases the k -value. Fan designation Ø represents a calm day and II approximately represents a day with the mean annual wind speed. A wider air space generally gives rise to lower k -value. If a dust strip is fitted between the casements, this is equivalent to the air space between casements being taped up. The table also shows that an increase in the width of gap between frame and casement on the outside initially produces a relatively large change in the k -value. As the width is further increased, there is hardly any change in the k -value. The large difference between a taped up air space and an air space width of 2.4 mm for fan designation Ø is also remarkable; the increase in the k -value is $0.11 \text{ W/m}^2\text{K}$. The reason for this is discussed below.

FIG.6.6.2-6.6.3 set out some temperature profiles for the air space between the bottom rails of the frame and casement. Temperatures have been measured at points on the outside of the frame, the outside of the casement, the rainwater channel, the air space between frame and inner casement, inside face of frame and inside face of casement. The profiles are largely the same irrespective of variations in outside air temperature, wind speed and air space width. The only case where there is a pronounced difference is when the air space between casements was sealed off (with a tape or duststrip) and the fan in the cold room was turned off (FIG.6.6.3). This implies that when the fan is turned on or when the air space between casements is open, cold air is drawn into the air space. When no force acts on the air, either forced convection due to the fans or natural convection between the panes ("chimney effect"), the air in the space at the bottom rail is practically stationary. Owing to the resulting higher temperature in the air space, heat flow at the bottom rail is reduced and the k -value decreases. TAB.6.6.2 sets out the temperatures in the rainwater channel in schematic groups. This

illustrates the state of affairs at the bottom. In the case of the stiles and the top rail the chimney effect does not come into play, but the fans must be turned on in order to reduce the temperature in the rainwater channel.

On the basis of these temperatures, an attempt has been made to predict the resulting value of λ in the gap between frame and casement. The value of thermal conductivity obtained in these approximate calculations for the inner part of the gap is of the order $\lambda=0.10-0.14$ W/mK, and it is practically independent of conditions in the cold room.

For the outer part of the gap, the value obtained in most cases was $\lambda=0.17-0.20$ W/mK. The only exception occurred when there is no fan to draw cold air into the outer part of the air space and rainwater channel. And for the air space at the bottom rail when the air space between casements is sealed and no fan is operating in the cold room. The value of λ for the outer part of the air space is then the same as in the inner part, i.e. $\lambda=0.10-0.14$ W/mK.

6.7 SURFACE COEFFICIENTS OF HEAT TRANSFER

6.7.1 Measurement of the coefficient on the inside of the glass

The aim of these measurements was to study the internal surface coefficients of heat transfer in a room. Heat flows were measured with heat flow sensors of 100 mm diameter and about 3 mm thickness by the auxiliary wall principle.

The heat flow sensors were checked by mounting them on a construction comprising 50 mm mineral wool insulation between two 4 mm thick sheets of glass. The thermal conductivity of the insulation had previously been determined in a Lang apparatus. The flow measured by the sensors was compared with the calculated value. It was found that the measured flows were 10-20% less than those calculated. The reason for this has been discussed in Subsection 5.4.2. The calibration factors of the heat flow sensors were therefore adjusted on the basis of the measurements.

Measurements were carried out on two different double glazed coupled outward opening windows. The air space between panes was in both casements 35 mm, the average distance between casements was 1.5-2 mm without a dust strip, and there was a P-shaped weatherstrip between inner casement and frame. The two windows were installed in a northerly and easterly facade. For one of the windows measurements were made with the air space between casements taped up. It should be noted that the air temperature at the level $x/L=0.50$ (FIG.6.7.1) is about the same in the room as in the vicinity of the window. The temperature gradient in the air in front of the window is approximately 1-1.5 °C/m. At the bottom of the reveal the cold downward air current reduces air temperature by about 1.5-2 °C. The measured values of the resistance of the air space (m_{sp}) and the inner surface coefficient of heat transfer (α) are set out in APP.6.7.2. The resistance of the air space was determined as the vertical temperature difference across the air space at level x divided by the corrected heat flow density ($q(x,z)$ W/m²), i.e.

$$m_{sp} = (\vartheta_{sp+}(x) - \vartheta_{sp-}(x))/q(x,z)$$

The inner surface resistance is calculated as

$$\alpha_{is} = q(x,z)/(\vartheta_i - \vartheta_{is}(x))$$

where

ϑ_i = air temperature perpendicular to the window in the room

$\vartheta_{is}(x)$ = surface temperature on the inside at level x

If the temperature ϑ_i is replaced by the temperature at about level x on the internal wall surface, there is no appreciable difference. The resistance of the air space is relatively constant in the vertical direction. The mean value for the two windows is almost the same. The vertical distribution of heat flow is changed greatest along the centre line, which is natural since convection along the vertical edges is obstructed. Taping up the air space increases thermal resistance by about 0.006 m²K/W, and this alters the thermal transmittance (k) of the window by

0.04 W/m²K (-1.6%).

The curves for the internal surface coefficient of heat transfer may be said to be mirror images of the thermal resistance of the air space. This is not so strange considering that heat flow is included in both the numerator and denominator of the expressions for m_{sp} and α_i . For window 1 α_i varies between 7.3 and 8.3 W/m²K, while for window 2 α_i = 6.8-9.9 W/m²K. The mean value of α_i is 7.5-8.1 W/m²K. Theoretical calculations according to Section 4.2, where

$$\bar{\alpha}_C + \bar{\alpha}_R = 2.2(T_i - T_{is})^{1/4} + \sigma \epsilon_{RES} 4T_m^3$$

yield a value between 8.4 and 8.8 W/m²K.

6.7.2 Surface coefficients of heat transfer for test windows P1-P6

The flows and temperatures obtained during measurements on test windows P1-P6 have been used in calculating the surface coefficients of heat transfer. These have been calculated as the density of heat flow rate through the surface (q) divided by the difference between the air temperature in the centre of the box (ϑ_i) and the surface temperature (ϑ_{is}), i.e.

$$\alpha_i(x) = \frac{q(x)}{\vartheta_i - \vartheta_{is}(x)}$$

For test windows P1-P4 the internal surface coefficients of heat transfer have been determined, see Jonsson (1985). For test window P6 the general appearance of α_i is shown in FIG.6.7.2. For the upper part α_i decreases in direct proportion to the distance from the upper edge. Inside the box there is a current of warm air which is cooled when it flows down the inside of the glass. When a convection movement is initiated, there is large surface transfer of heat, i.e. the value of α_i is high. The lower part of the window is affected mostly by the convection inside the window. Generally, the value of α_i varies from about 11 to about 7 W/m²K between the top and the bottom.

Measurements relating to two and three air spaces have also been plotted in this figure. These have the same pattern as the curves for one air space.

TAB.6.6.1 K-values measured at a) variable outside temperatures,
 b) fan arrangements and c) air space widths (with
 $m_i + m_u = 0.20 \text{ m}^2 \text{K/W}$).

a)

	0 °C	-10 °C	-25 °C
Fan II, $d_1=2.4 \quad d_2=2.8$	1.65	1.65	1.66
Fan II, $d_1=7.0 \quad d_2=9.4$	1.77	1.74	1.74

b)

	Fan Ø	Fan I	Fan II
$\vartheta_u \sim -10 \text{ °C}$ $d_1=2.4 \quad d_2=2.8$	1.62	1.64	1.66

c)

	$d_1=2.4$ $d_2=2.8$ taped	$d_1=2.4$ $d_2=2.8$	$d_1=5.0$ $d_2=2.8$	$d_1=7.0$ $d_2=2.8$	$d_1=7.0$ $d_2=5.8$	$d_1=7.0$ $d_2=9.4$
Fan II, $u -10 \text{ °C}$	1.61	1.65	1.67	1.67	1.74	1.74
Fan Ø, $u -10 \text{ °C}$	1.51	1.62	1.65	1.68	1.71	1.72
Fan II, $u -10 \text{ °C}$ Dust strip	1.61	1.61	1.62	1.62	-	-
Fan Ø, $u -10 \text{ °C}$ Dust strip	1.51	1.53	1.53	1.52	-	-

TAB.6.6.2 Temperatures in rainwater channel at top rail, stiles and bottom rail.

Case		Window frame head	Jamb		Sill
fan	d_1		hinged side		
II	0-7.0	0.10-0.21	0.10-0.19	0.07-0.14	0.12-0.18
Ø	2.4-7.0	0.44-0.49	0.26-0.32	0.28-0.31	0.13-0.16
Ø	0	0.42-0.44	0.38	0.29	0.33-0.36

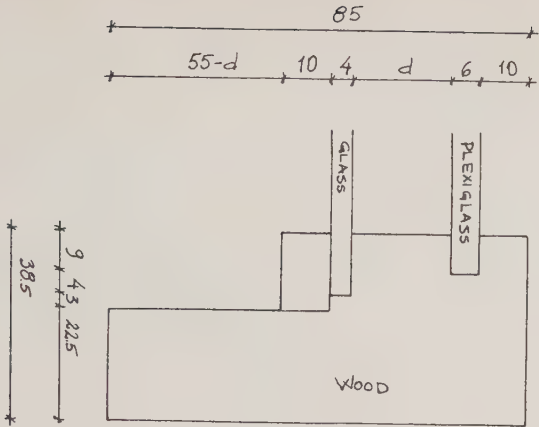


FIG.6.1.1 Profile of test windows P1, P2, P3 and P4. Outside dimensions 1200x1200 mm, 918x1200 mm, 638x1200 mm and 349x1200 mm, size of air space 1123x1123 mm, 841x1123 mm, 561x1123 mm and 272x1123 mm. Dimensions given are heightxwidth (Lxb) according to Subsection 2.3.3.2.

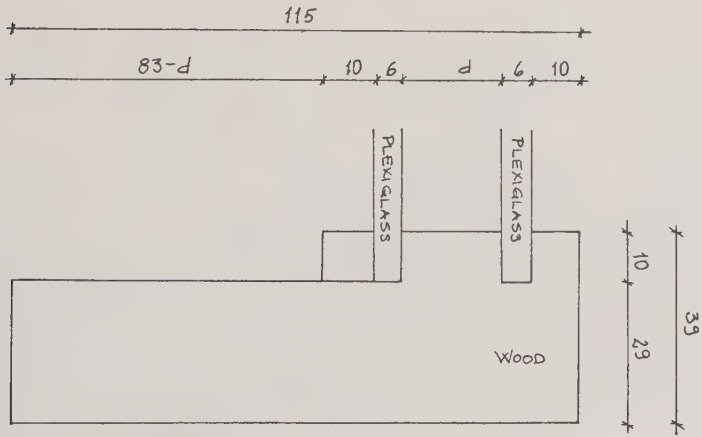


FIG.6.1.2 Profile of test window P6. Outside dimensions 918x1200 mm, air space 840x1122 mm (heightxwidth).

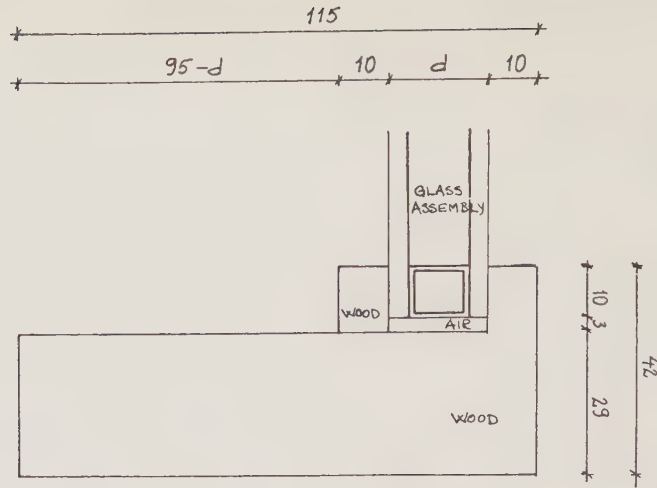


FIG.6.1.3 Profile of test window P7. Outside dimensions 918x1200 mm, air space 834x1116 mm, sealed unit 855x1137 mm (heightxwidth).

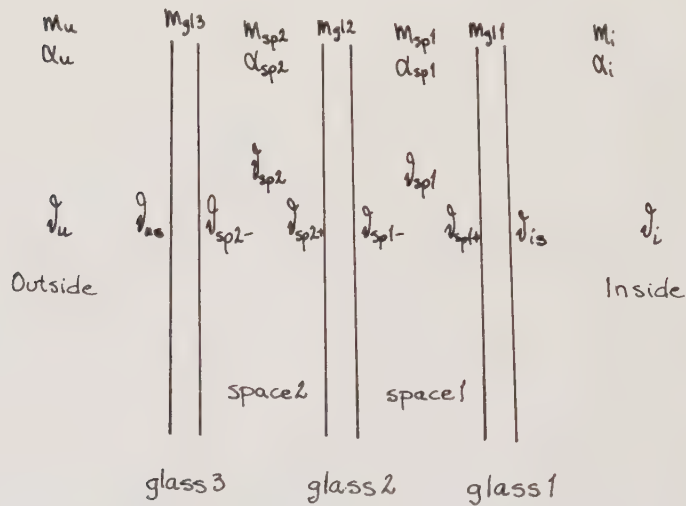


FIG.6.1.4 Symbols used for windows. m = thermal resistance, α = surface coefficient of heat transfer, T = temperatures, $sp1 \dots spn$ = space 1 ..n, i and u = inside and outside respectively, s = surface, $+-$ = warm and cold side of air space respectively. Numbering begins on the warm side.

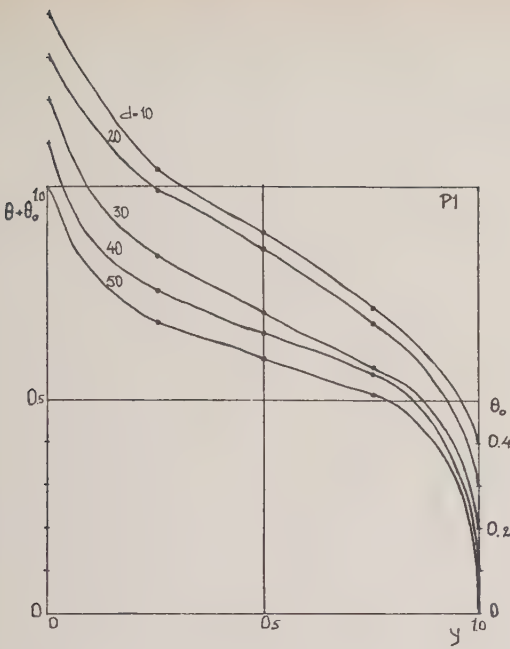


FIG.6.2.1

Horizontal temperature distribution in the air space. $x_1/L=0.38$, $L=1125$ mm, $d=10, 20, 30, 40$ and 50 mm, $L/d=112, 56, 37, 28$ and 22 , $Ra=2900, 22600, 77500, 190500$ and 366000 .

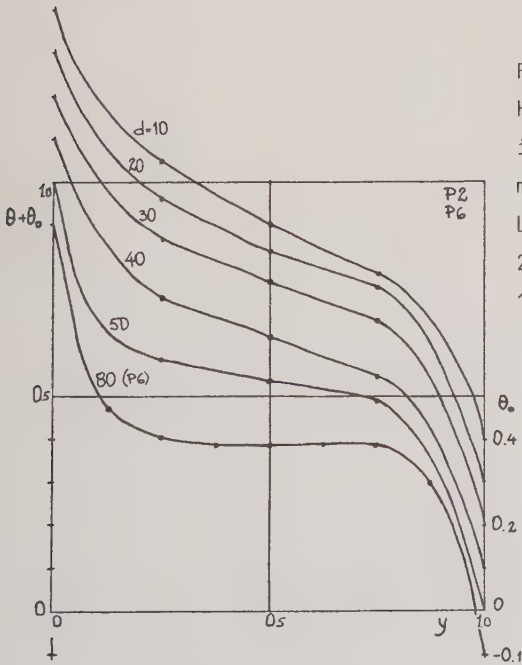


FIG.6.2.2

Horizontal temperature distribution in the air space. $x_1/L=0.50$, $L=841$ mm, $d=10, 20, 30, 40, 50$ mm and 80 mm, $L/d=84, 42, 28, 21, 17$ and 11 , $Ra=2900, 22900, 102600, 177300, 439100$ and $1.8 \cdot 10^6$.

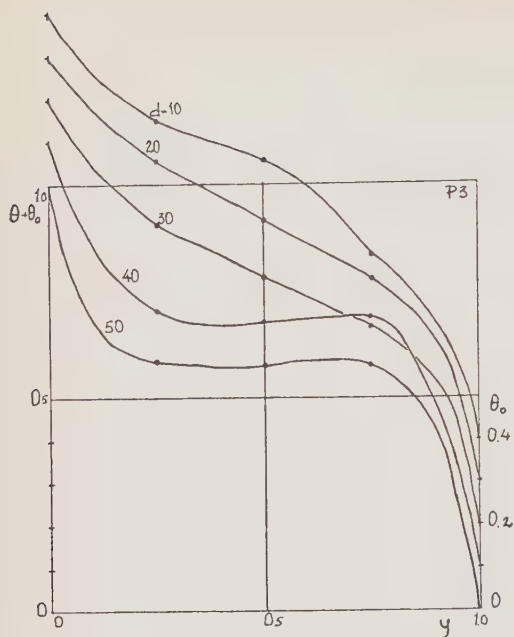


FIG.6.2.3

Horizontal temperature distribution in the air space. $x_1/L=0.44$, $L=561$ mm, $d=10, 20, 30, 40$ and 50 mm, $L/d=56, 28, 19, 14$ and 11 , $Ra=3000, 22900, 108300, 263500$ and 369900 .

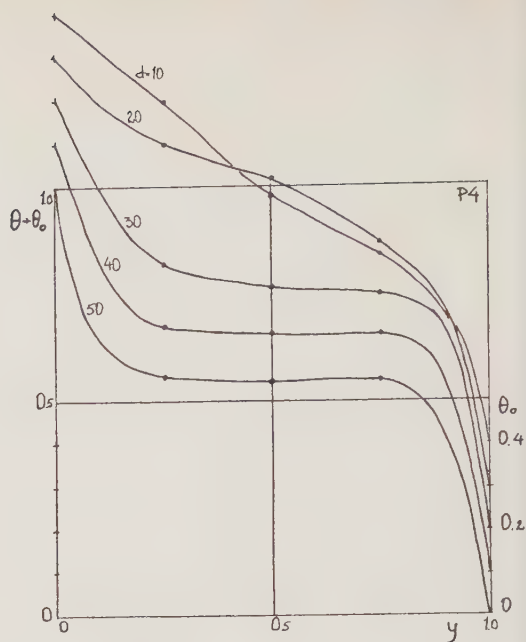


FIG.6.2.4

Horizontal temperature distribution in the air space. $x_1/L=0.60$, $L=272$ mm, $d=10, 20, 30, 40$ and 50 mm, $L/d=27, 14, 9, 7$ and 5 , $Ra=2900, 22600, 77200, 181900$ and 335700 .

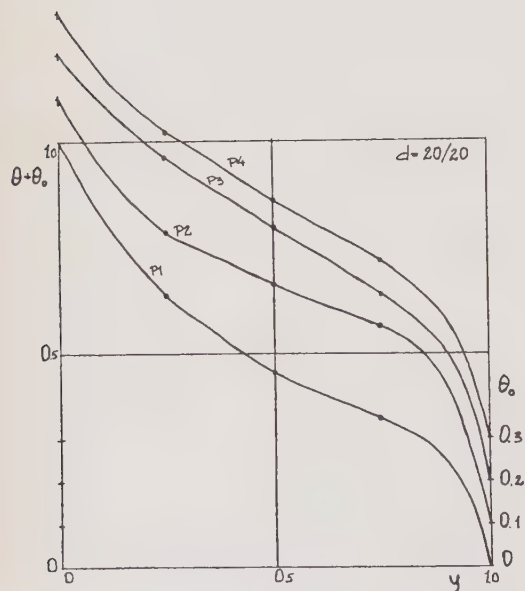


FIG.6.2.5

Horizontal temperature distribution in the warm air space in test windows P1, P2, P3 and P4 with two air spaces, $20+20$ mm. $L/d=56, 42, 28$ and 14 , $Ra=14900, 12400, 13600$ and 11800 .

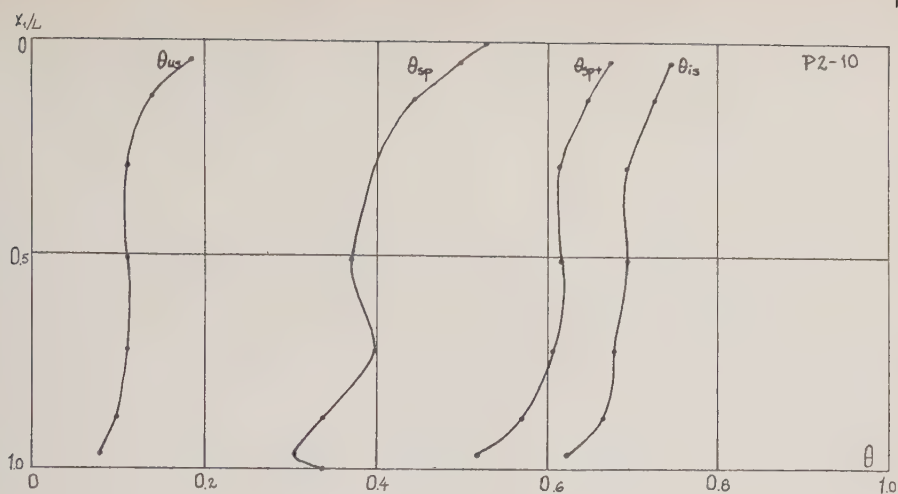


FIG.6.2.6 Vertical temperature distribution on the cold surface, in the middle of the air space and on the warm inner surfaces. $L=841$ mm, $d=10$ mm and $Ra=2900$.

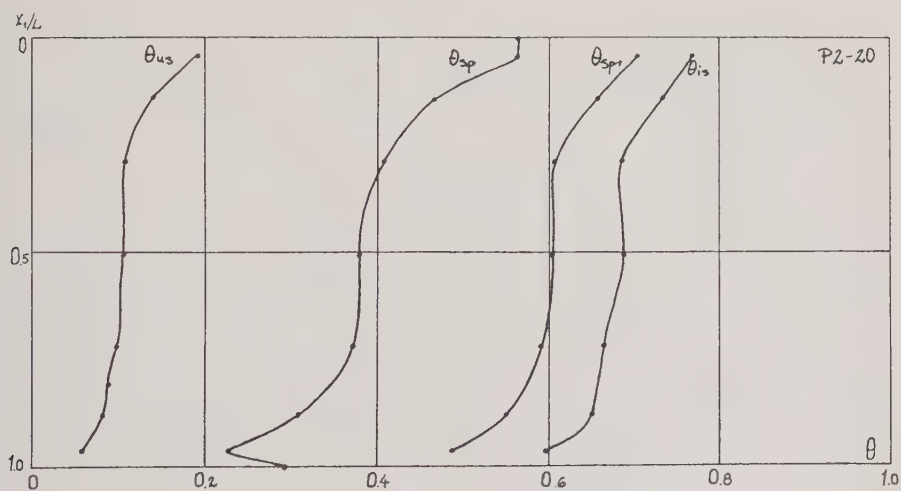


FIG.6.2.7 Vertical temperature distribution on the cold surface, in the middle of the air space and on the warm inner surfaces. $L=841$ mm, $d=20$ mm and $Ra=22900$.

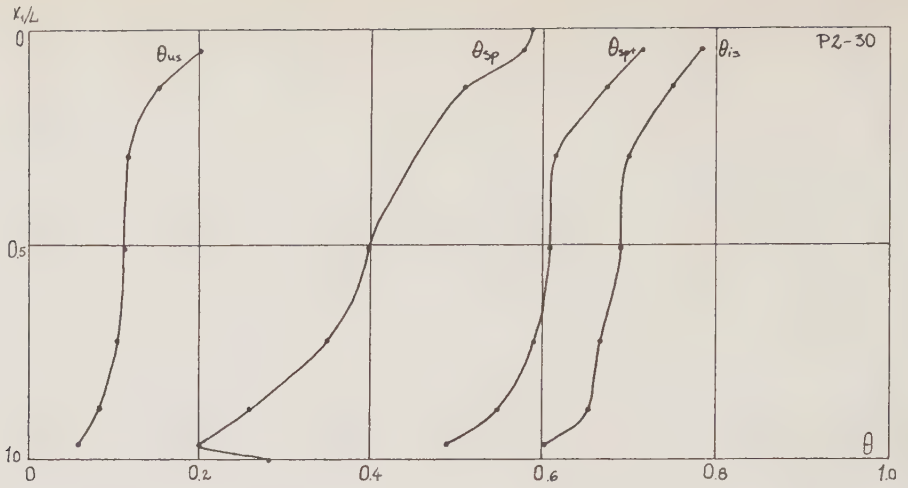


FIG.6.2.8 Vertical temperature distribution on the cold surface, in the middle of the air space and on the warm inner surfaces. $L=841$ mm, $d=30$ mm and $Ra=102600$.

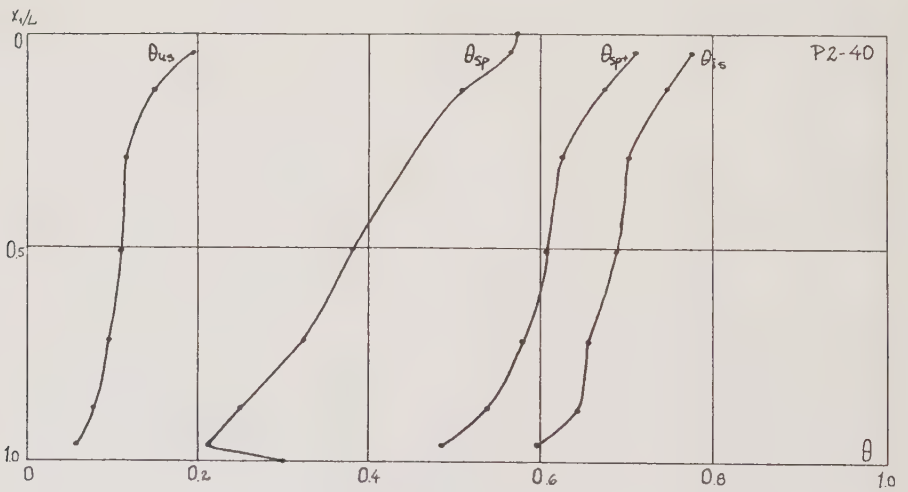


FIG.6.2.9 Vertical temperature distribution on the cold surface, in the middle of the air space and on the warm inner surfaces. $L=841$ mm, $d=40$ mm and $Ra=177300$.

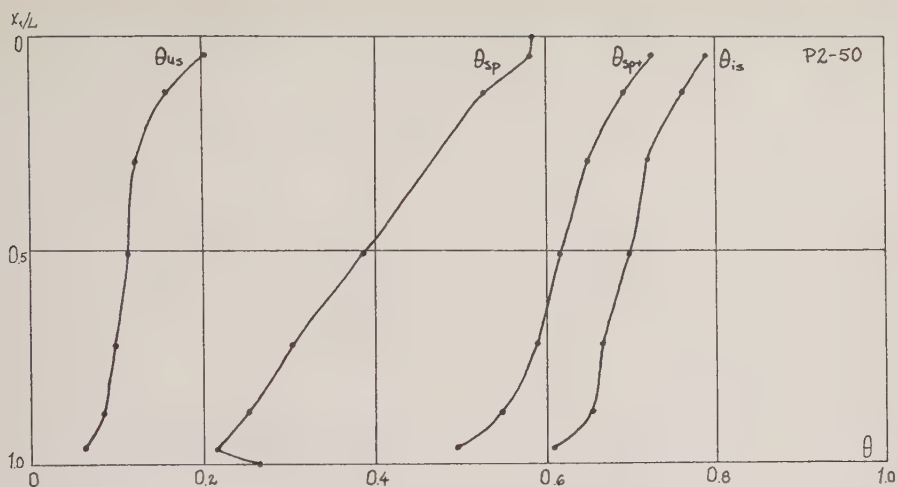


FIG.6.2.10 Vertical temperature distribution on the cold surface, in the middle of the air space and on the warm inner surfaces. $L=841$ mm, $d=50$ mm and $Ra=493100$.

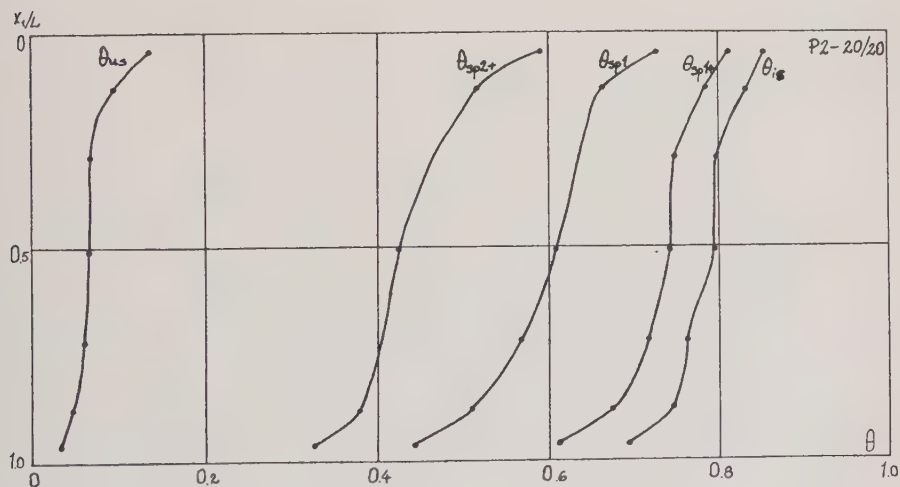


FIG.6.2.11 Vertical temperature distribution on the cold surface, the middle pane, in the middle of the warm air space and on the warm inner surfaces in test window P2 with two air spaces, $20+20$ mm. $Ra=12400$.

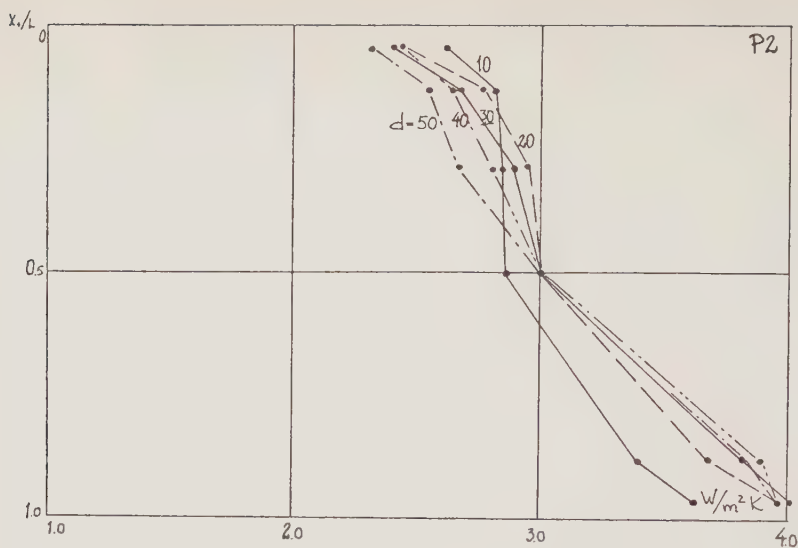


FIG.6.2.12 Total flow along the inner surface divided by the indoor air-outdoor air temperature difference. $L=841$ mm, $d=10, 20, 30, 40$ and 50 mm.

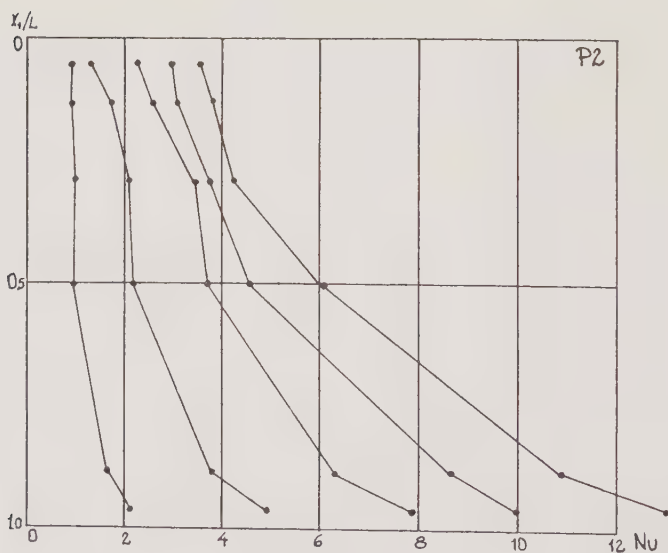


FIG.6.2.13 The Nusselt number along the inner surface. $L=841$ mm, $d=10, 20, 30, 40$ and 50 mm.

m^2k/w

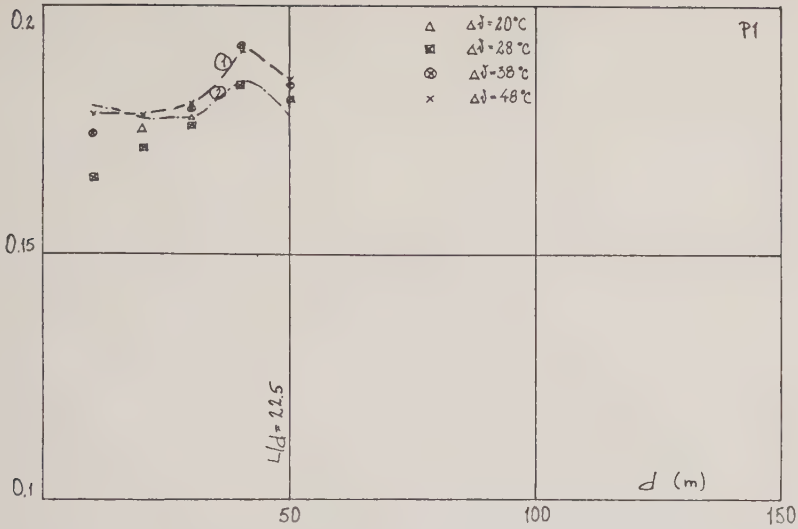


FIG.6.2.14 Thermal resistance as a function of the air space width for test window P1 according to measurements in the hot box. Curve 1 = air space + spacer, Curve 2 = air space. (Curve 1 corrected).

m^2k/w

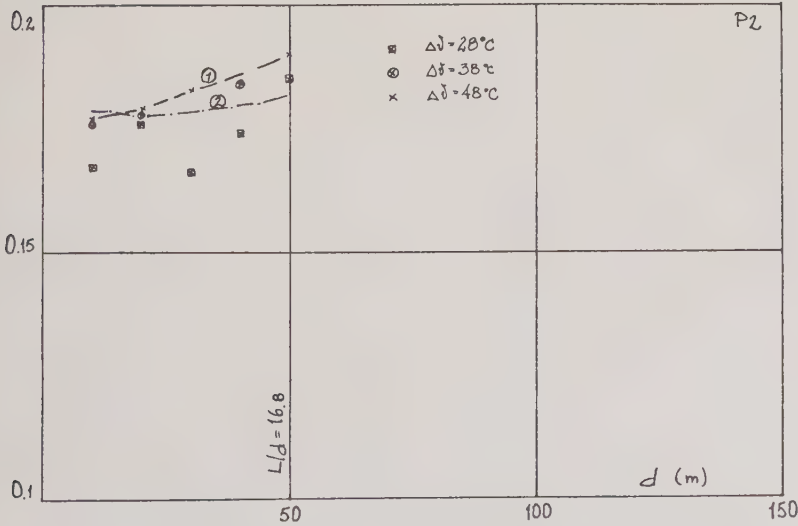


FIG.6.2.15 Thermal resistance as a function of the air space width for test window P2 according to measurements in the hot box. Curve 1 = air space + spacer, Curve 2 = air space. (Curve 1 corrected).

m^2K/W

P3

 m^2K/W

P4

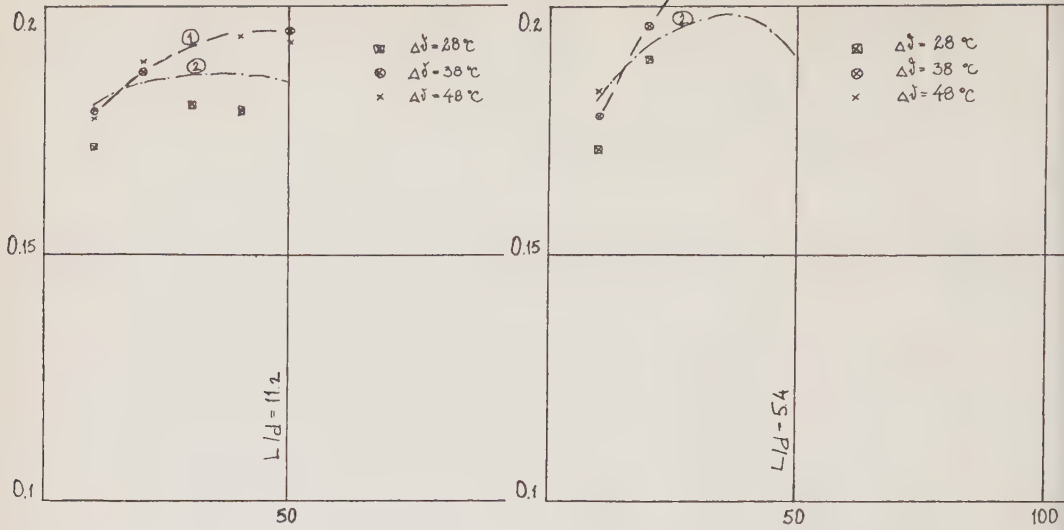


FIG.6.2.16-17 Thermal resistance as a function of the air space width for test window P3 and P4 according to measurements in the hot box. Curve 1 = air space + spacer, Curve 2 = air space. (Curve 1 corrected).

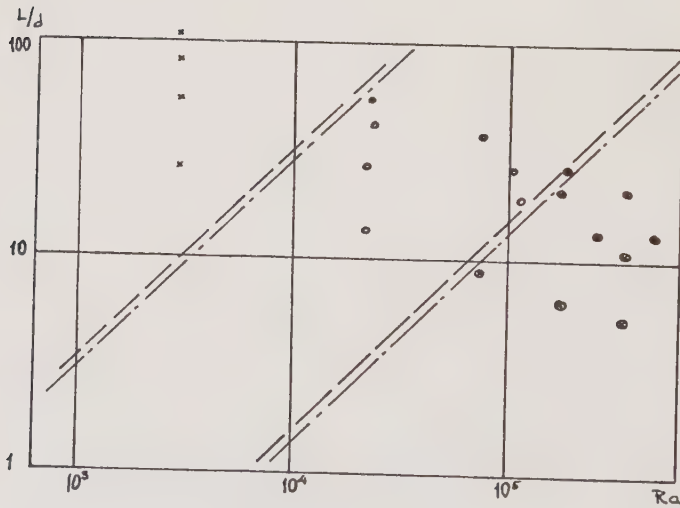


FIG.6.2.18 Classification into flow regimes on the basis of the height/width ratio (L/d) and the Rayleigh number (Ra). x conduction regime, o transition regime, ⊗ boundary layer regime, --- according to measurements for test windows P1, P2, P3 and P4 (see FIG. 6.2.1-6.2.4), —, according to Yin, Wung, Chen (1978).

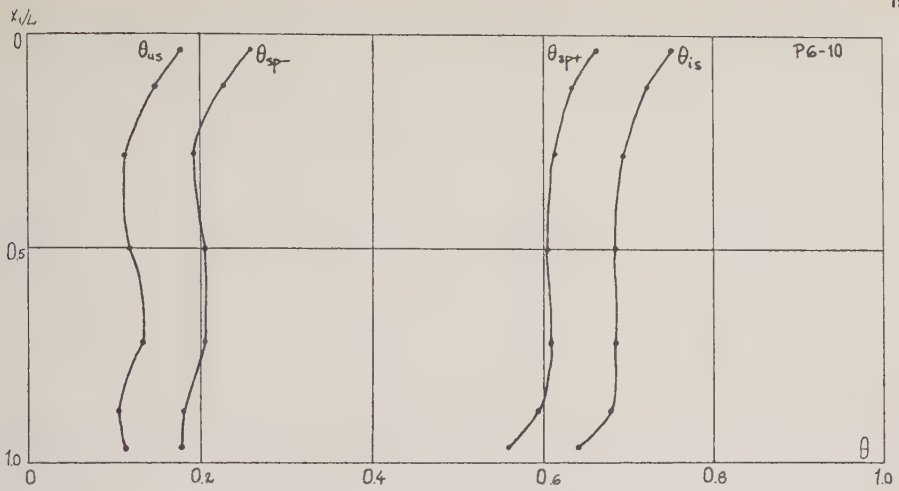


FIG.6.3.1 Vertical temperature distribution along the centre line for the outer and inner surfaces. One air space ($d=10$ mm).

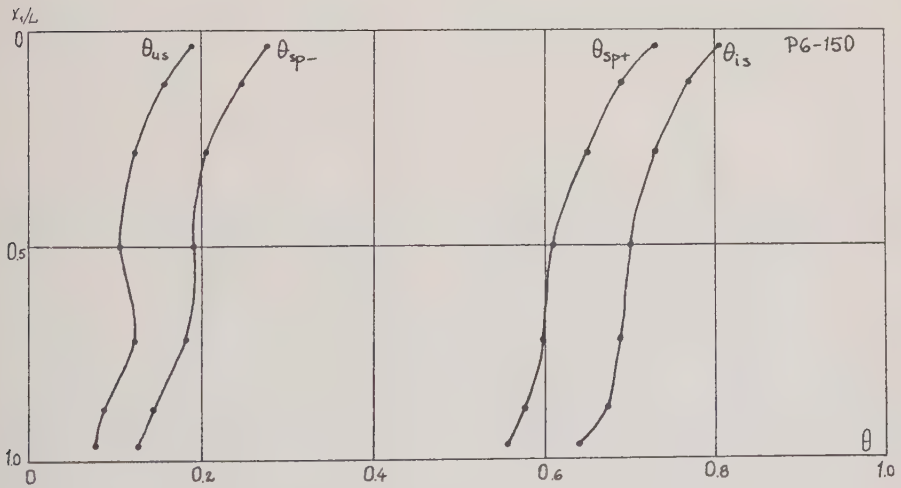


FIG.6.3.2 Vertical temperature distribution along the centre line for the outer and inner surfaces. One air space ($d=150$ mm).

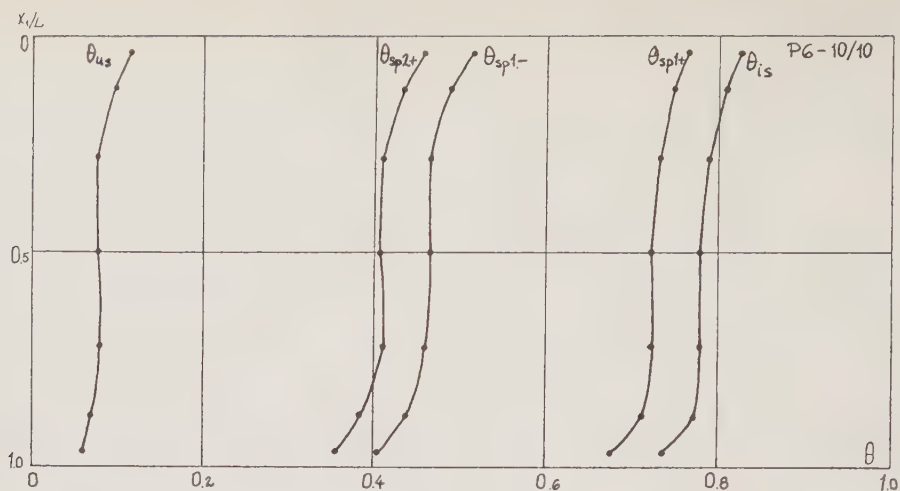


FIG.6.3.3 Vertical temperature distribution along the centre line for the outer cold surface and both surfaces of the intermediate and inner pane. Two air spaces ($d=10+10$ mm).

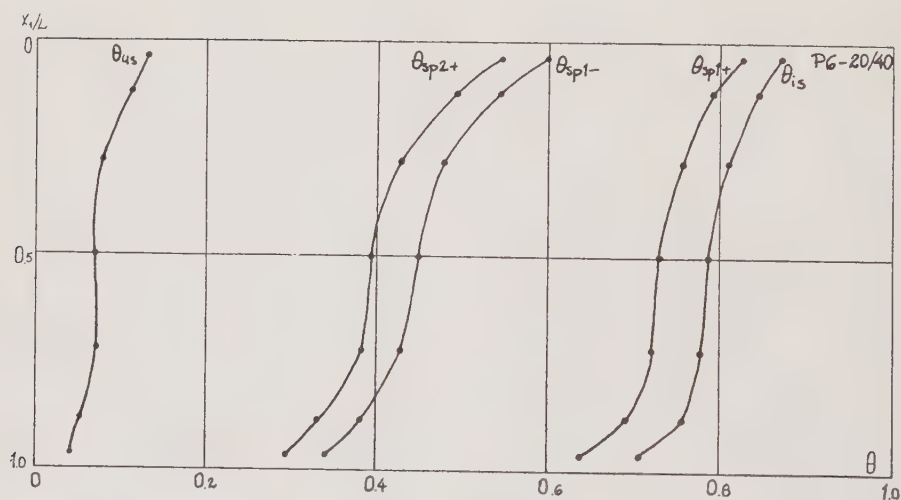


FIG.6.3.4 Vertical temperature distribution along the centre line for the outer cold surface and both surfaces of the intermediate and inner pane. Two air spaces ($d=20+40$ mm).

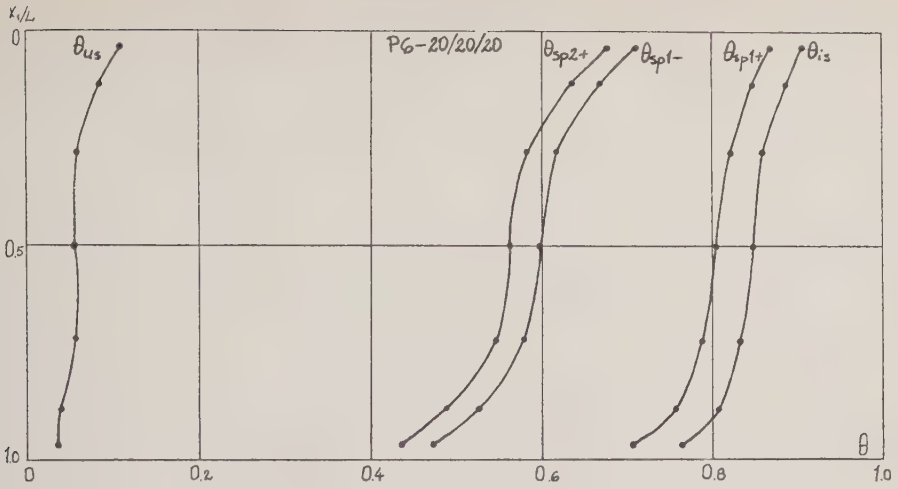


FIG.6.3.5 Vertical temperature distribution along the centre line for the outer cold surface and both surfaces of the intermediate and inner pane. Three air spaces ($d=20+20+20$ mm).

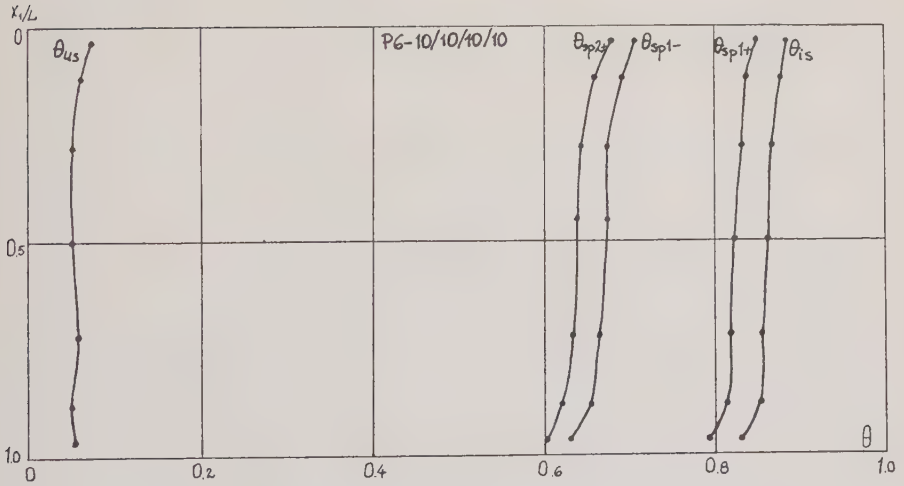


FIG.6.3.6 Vertical temperature distribution along the centre line for the outer cold surface and both surfaces of the intermediate and inner pane. Four air spaces ($d=10+10+10+10$ mm).

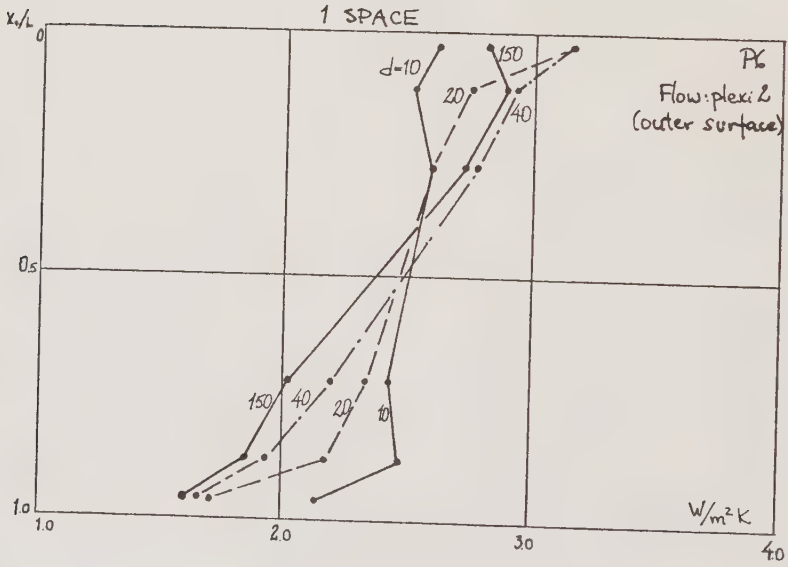
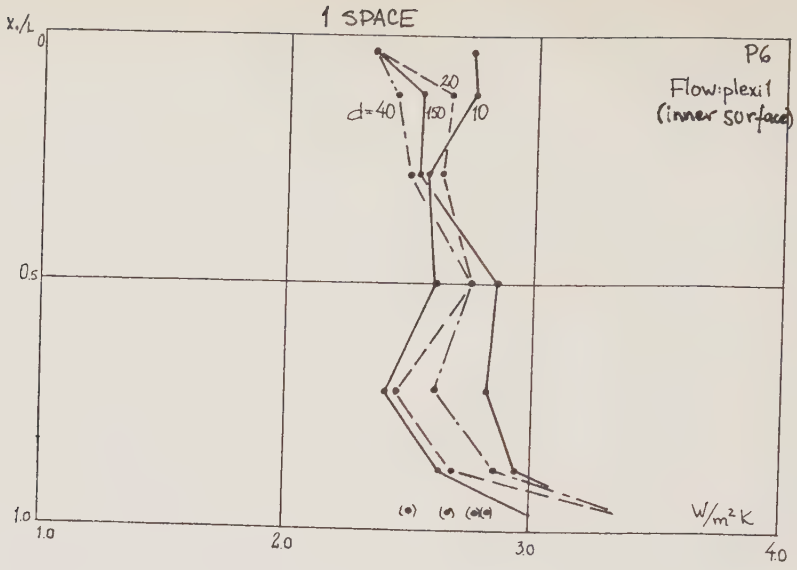


FIG.6.3.7 Heat flow divided by the indoor air-outdoor air temperature difference, calculated along the centre line on the inner and outer surface for air spaces $d=10, 20, 40$ and 150 mm.

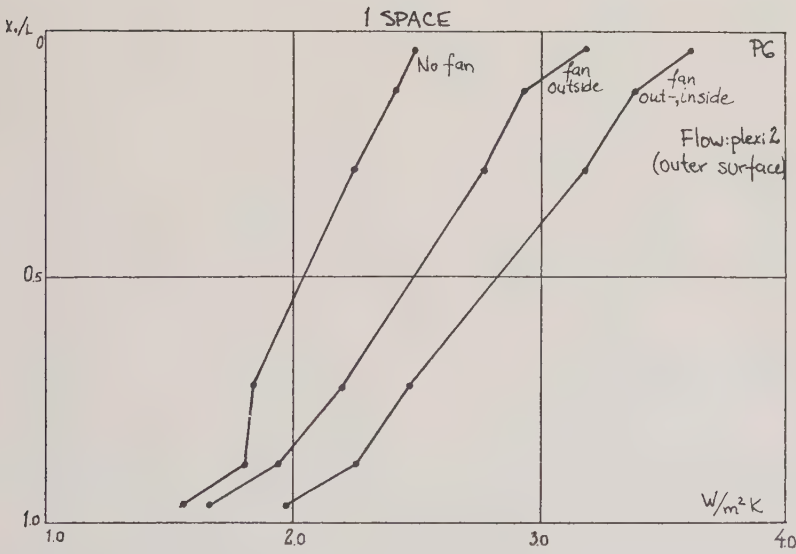
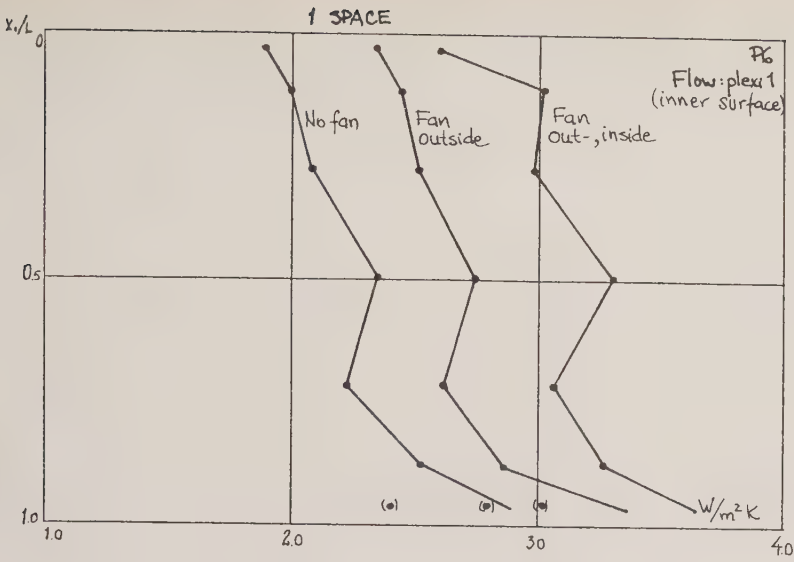


FIG.6.3.8 Heat flow divided by the indoor air-outdoor air temperature difference, calculated along the centre line on the inner and outer surface for air space $d=40$ mm. No fans, fans on the cold side and fans on both the cold and warm side.

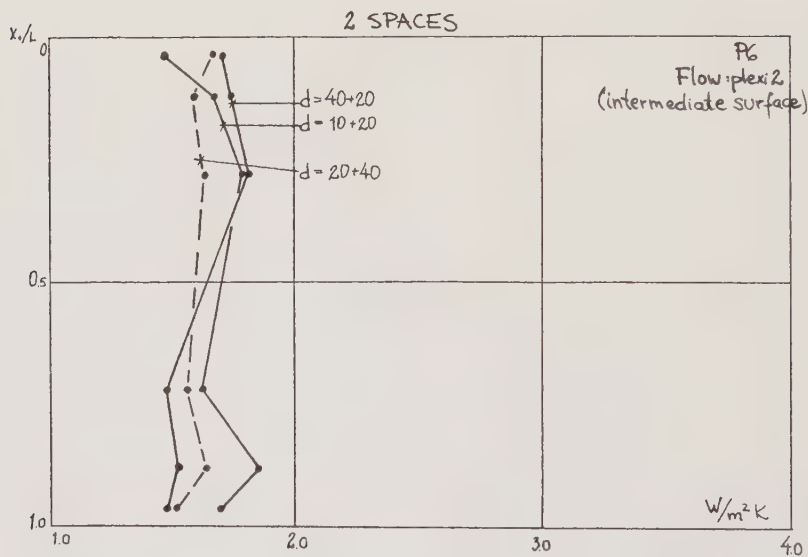
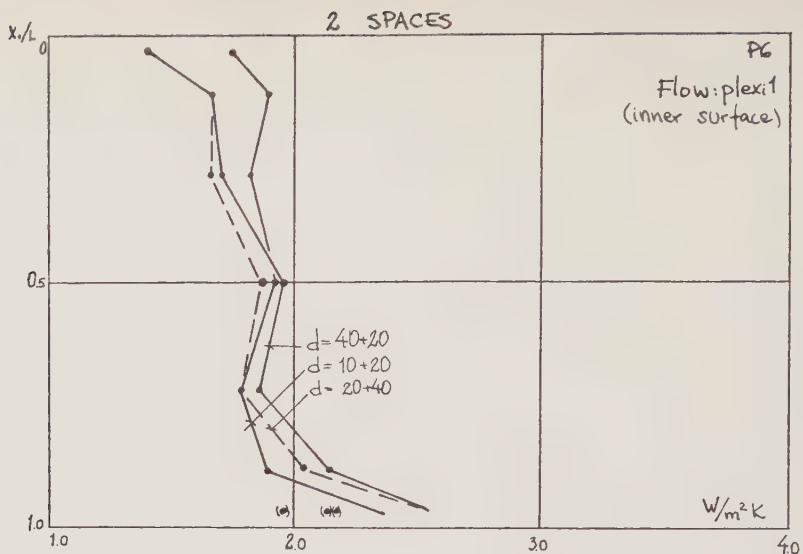


FIG.6.3.9 Heat flow divided by the indoor air-outdoor air temperature difference, calculated along the centre line on the inner surface for air space widths $d=10, 20$ and 40 mm, and on the next surface for air space widths $d=10, 20$ and 40 mm. Two air spaces ($d=10+20, 20+40$ and $40+20$ mm).

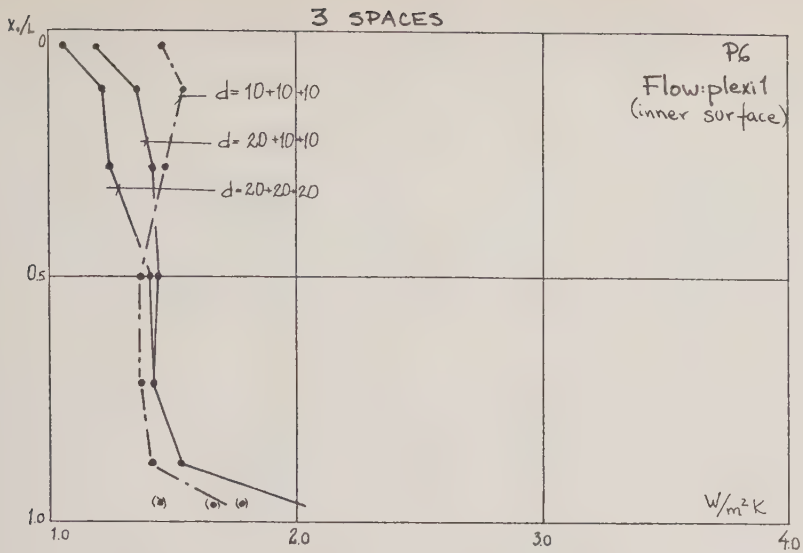


FIG.6.3.10 Heat flow divided by the indoor air-outdoor air temperature difference, calculated along the centre line on the inner surface for air space widths $d=10$ and 20 mm. Three air spaces ($d=10+10+10$, $20+10+10$, and $20+20+20$ mm).

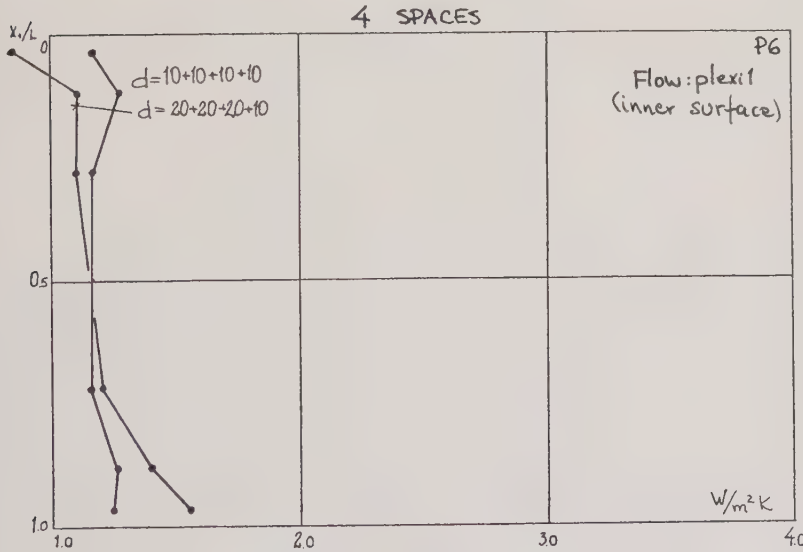


FIG.6.3.11 Heat flow divided by the indoor air-outdoor air temperature difference, calculated along the centre line on the inner surface for air space widths $d=10$ and 20 mm. Four air spaces ($d=10+10+10+10$ and $20+20+20+10$ mm).

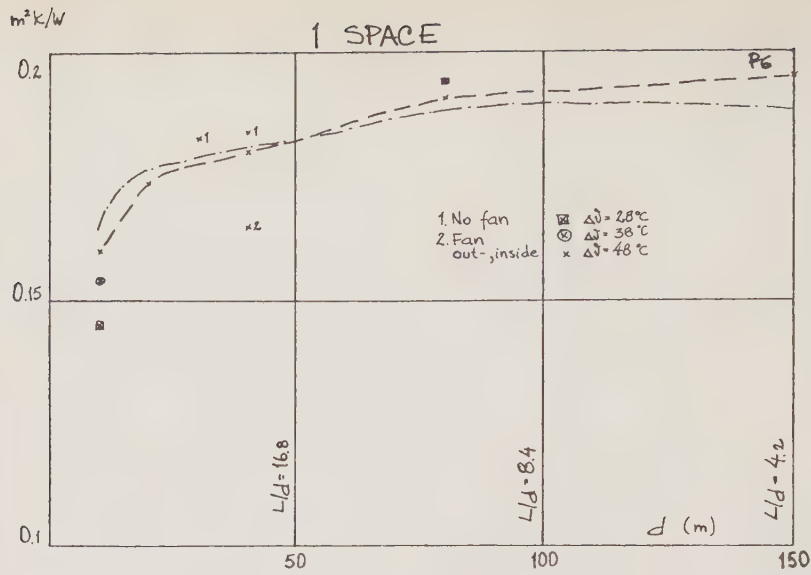


FIG.6.3.12 Thermal resistance according to measurements in the hot box for test window P6 (air space+spacer).
Curve -.- corrected for change in fabric loss through frame and wall according to Section 5.2.

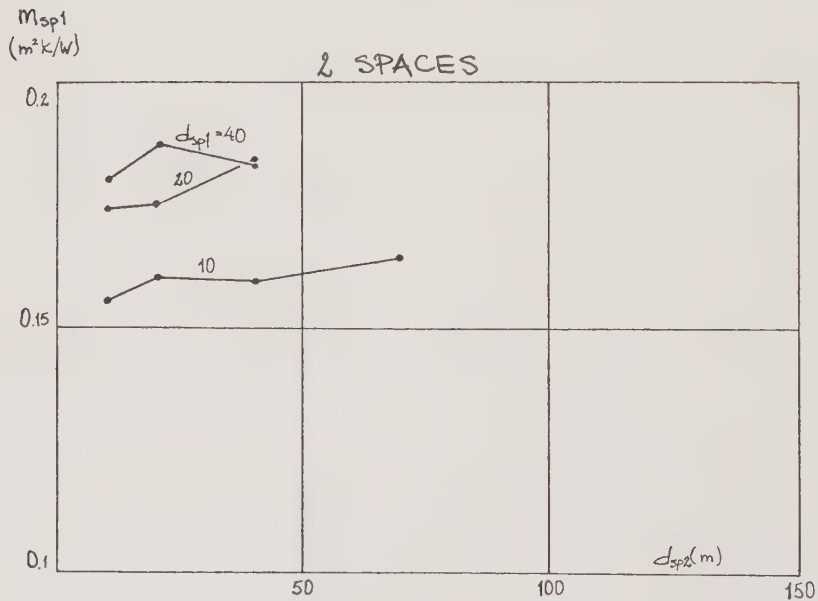


FIG.6.3.13 Thermal resistance according to measurements in the hot box for two air spaces (incl spacers). d_{sp1} and d_{sp2} = width of inner and outer air space.

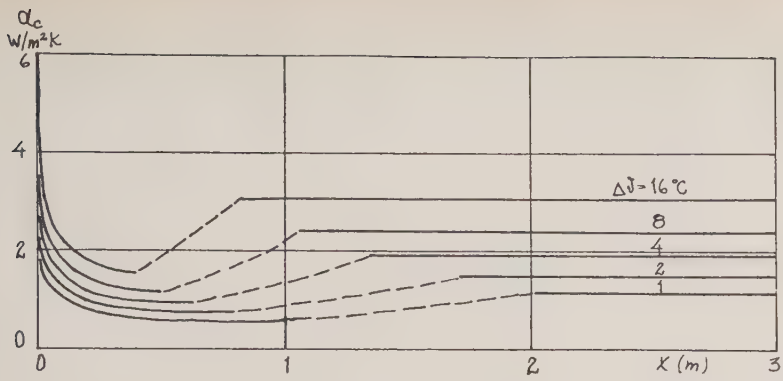


FIG.6.3.14 Variation in convective heat transfer as a function of the distance from the edge of the plate. According to Eckert (1959).

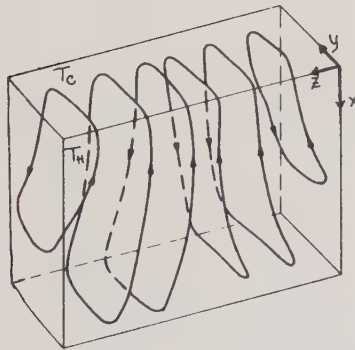


FIG.6.3.15 Structure of three dimensional flow according to Morrison, Iran (1978).

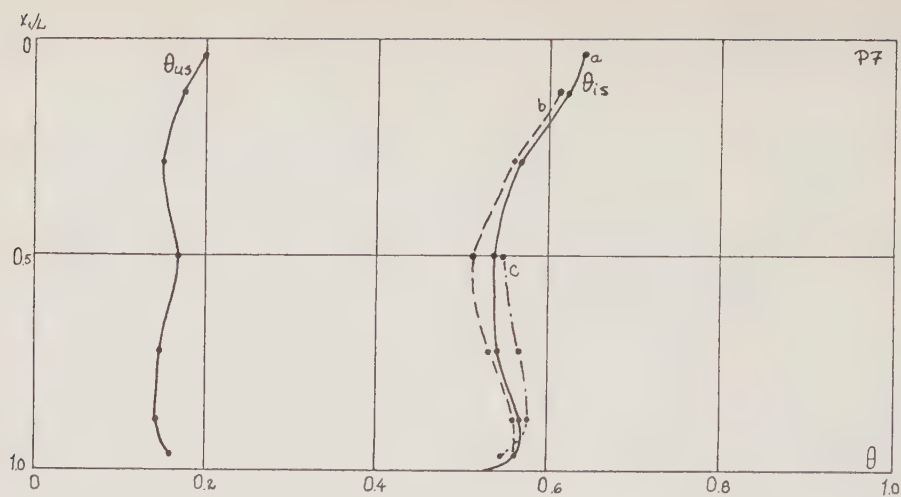


FIG.6.4.1 Vertical temperature distribution along
a) centre line $z=0.50$, b) $z=0.375$, and c) $z=0.25$.
D4-6.

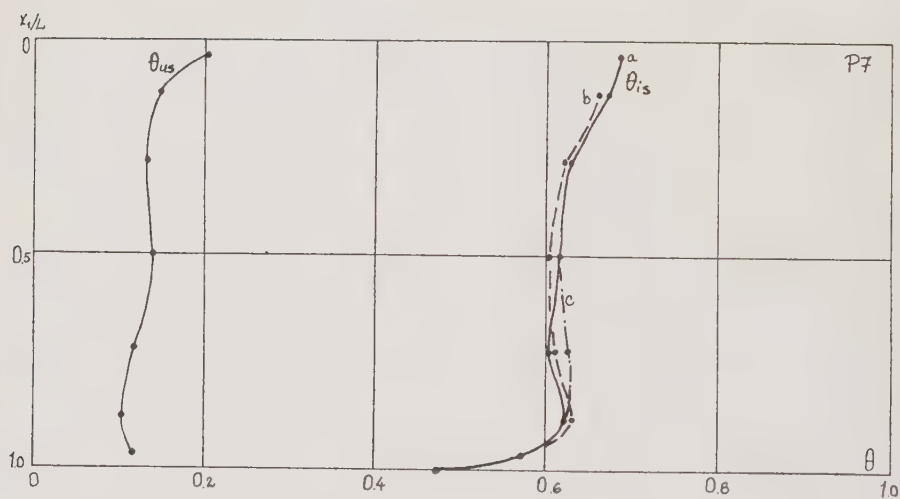
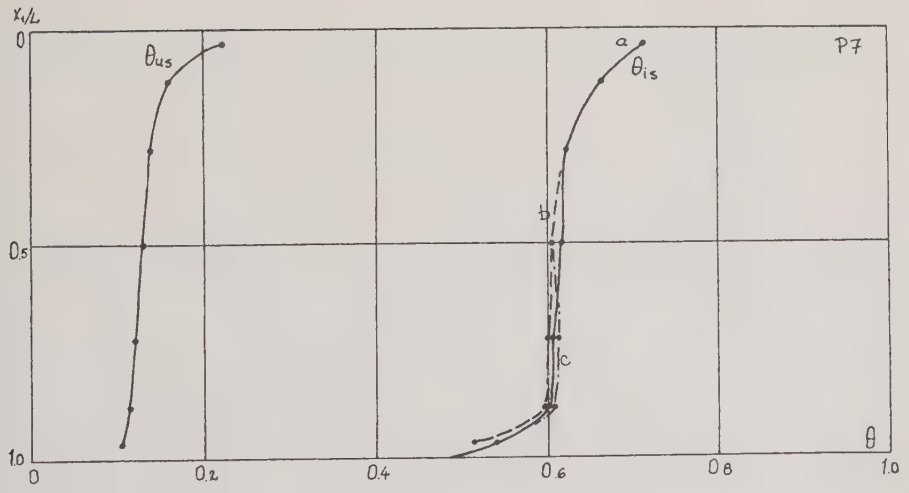
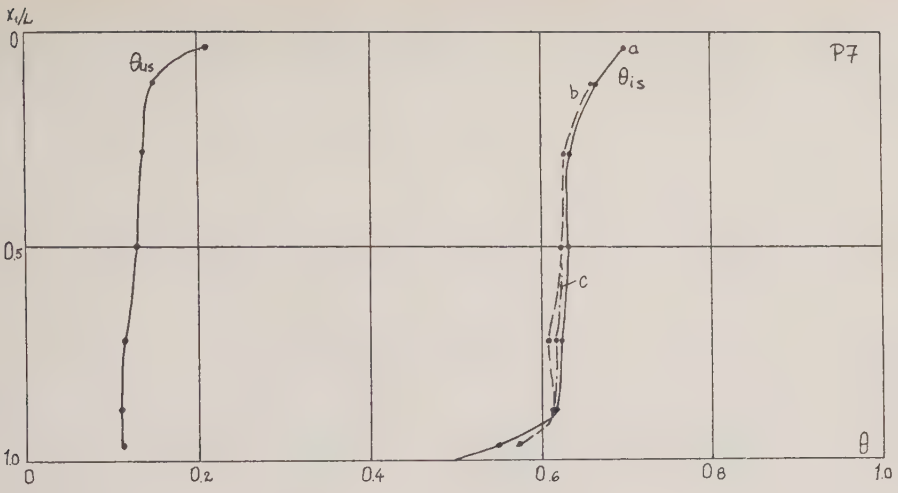


FIG.6.4.2 Vertical temperature distribution along
a) centre line $z=0.50$, b) $z=0.375$, and c) $z=0.25$.
D4-12.



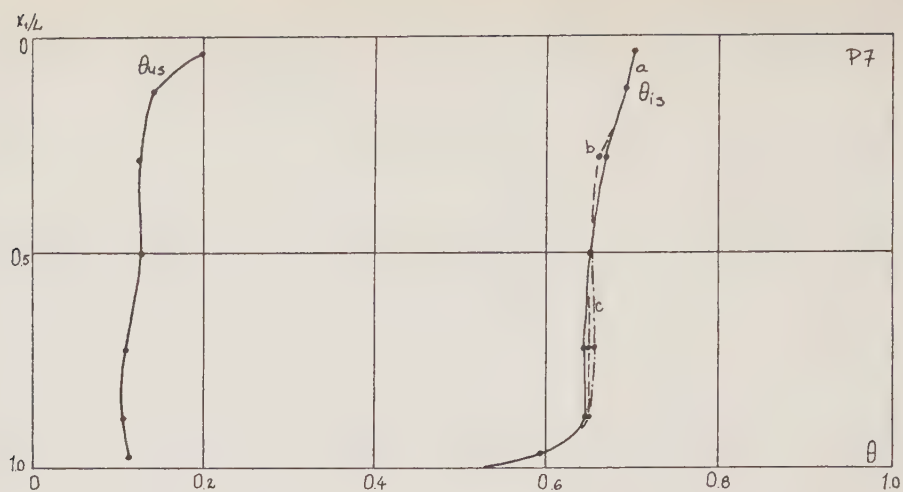


FIG.6.4.5 Vertical temperature distribution along
a) centre line $z=0.50$, b) $z=0.375$, and c) $z=0.25$.
D4-12, Ar.

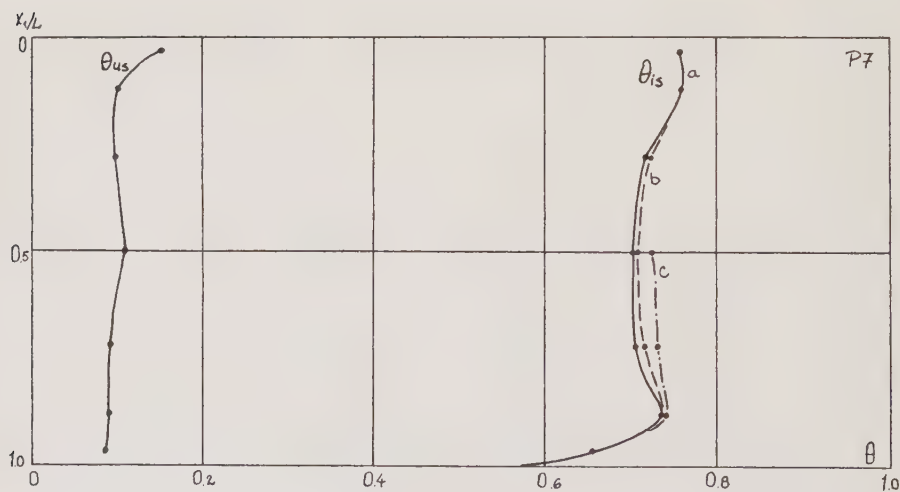


FIG.6.4.6 Vertical temperature distribution along
a) centre line $z=0.50$, b) $z=0.375$, and c) $z=0.25$.
D4-12, low emissivity layer.

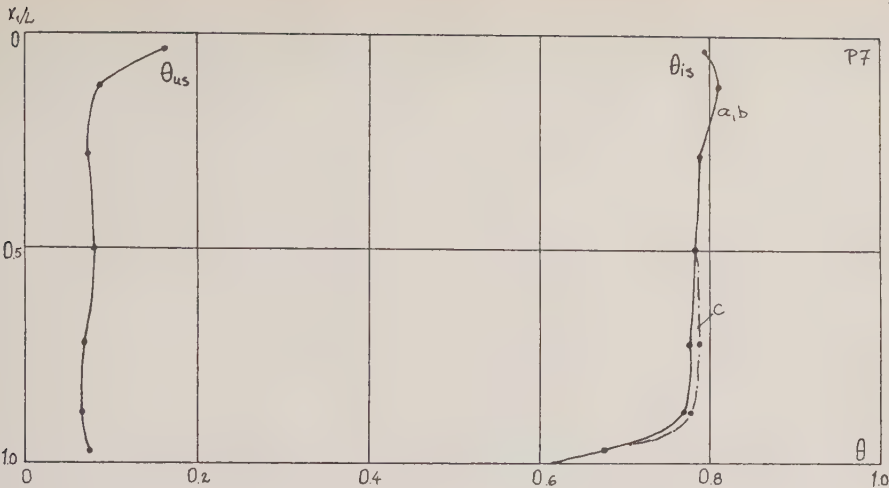


FIG.6.4.7 Vertical temperature distribution along
a) centre line $z=0.50$, b) $z=0.375$, and c) $z=0.25$.
D4-12, Ar, low emissivity layer.

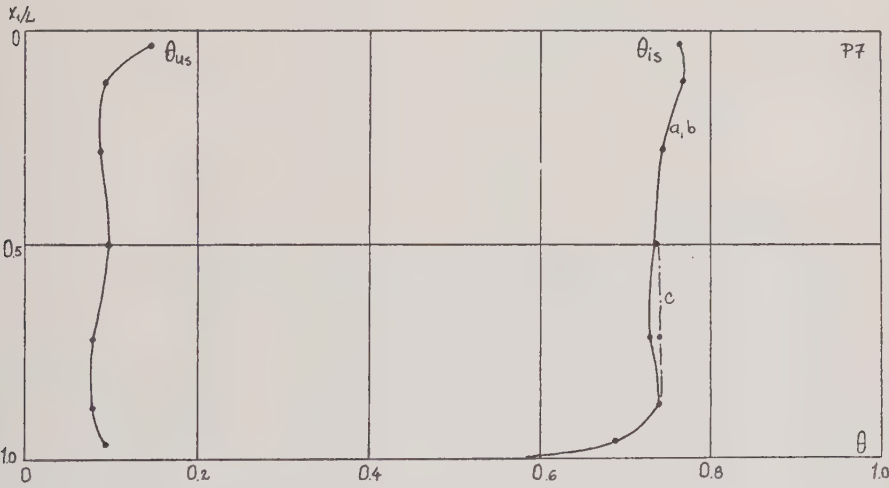


FIG.6.4.8 Vertical temperature distribution along
a) centre line $z=0.50$, b) $z=0.375$, and c) $z=0.25$.
T4-12.

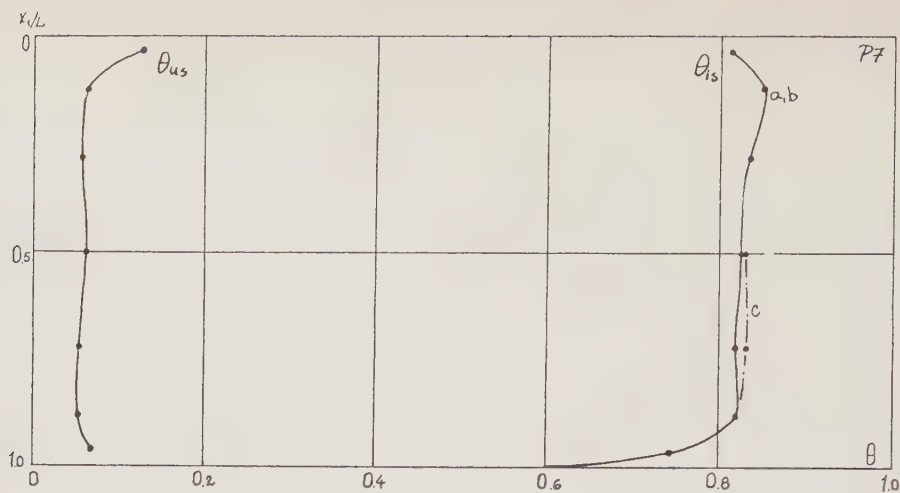


FIG.6.4.9 Vertical temperature distribution along
a) centre line $z=0.50$, b) $z=0.375$, and c) $z=0.25$.
T4-12, Ar, low emissivity layer.

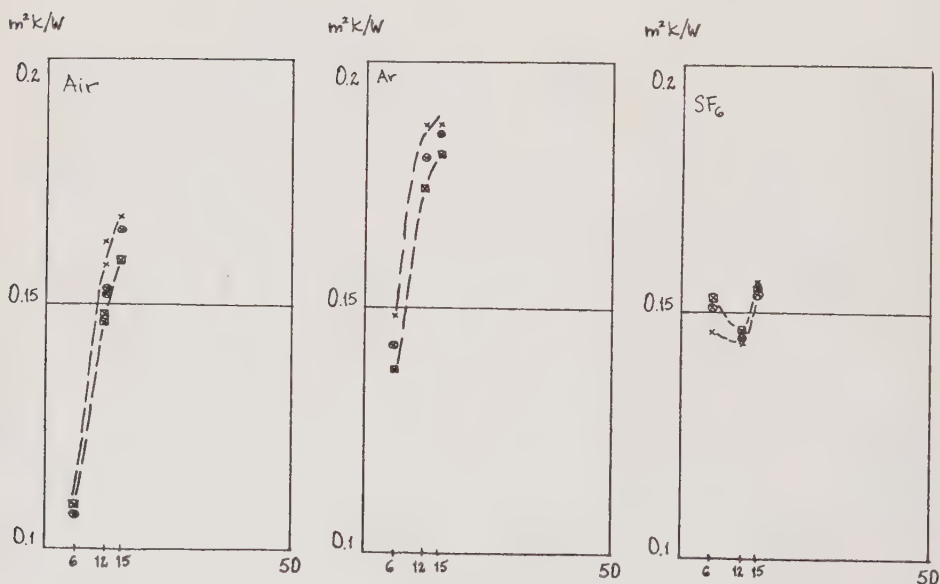


FIG.6.4.10 Thermal resistance (mean value for whole pane) for
spaces containing air, argon or SF_6 ($d=6-15$ mm),
according to measurements in the hot box.

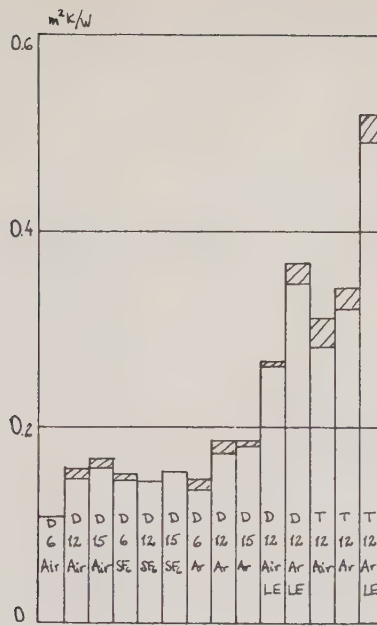


FIG.6.4.11 Thermal resistance (mean value for whole pane) according to measurements in the hot box for all the tested sealed units. The values shown in the cross hatched areas were obtained for different temperature differences.

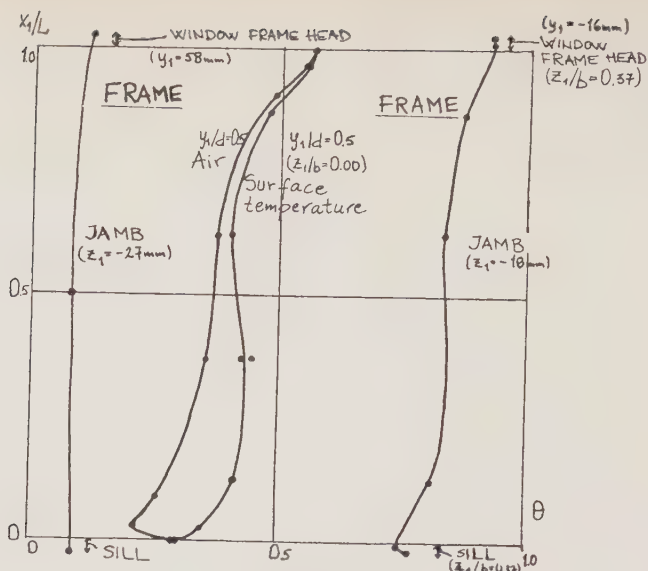


FIG.6.5.1 Nondimensional temperature for the frame abutting on the cold and warm room and surface temperature abutting on the air space. The air temperature refers to the vertical centre line.

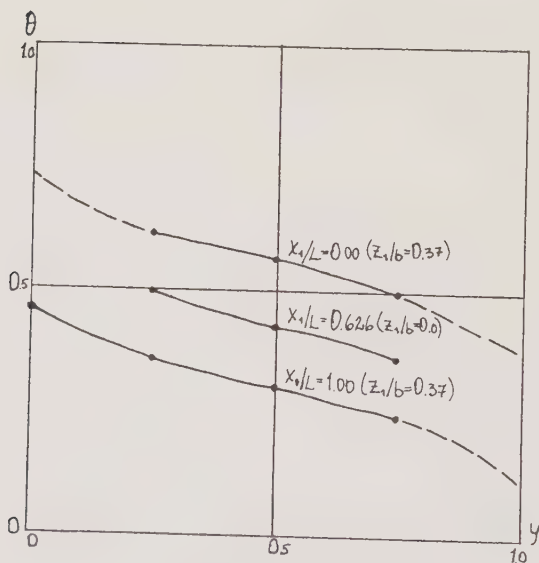


FIG.6.5.2 Nondimensional surface temperature for the frame abutting on the air space, for $x=0, 0.626$ and 1.0 .

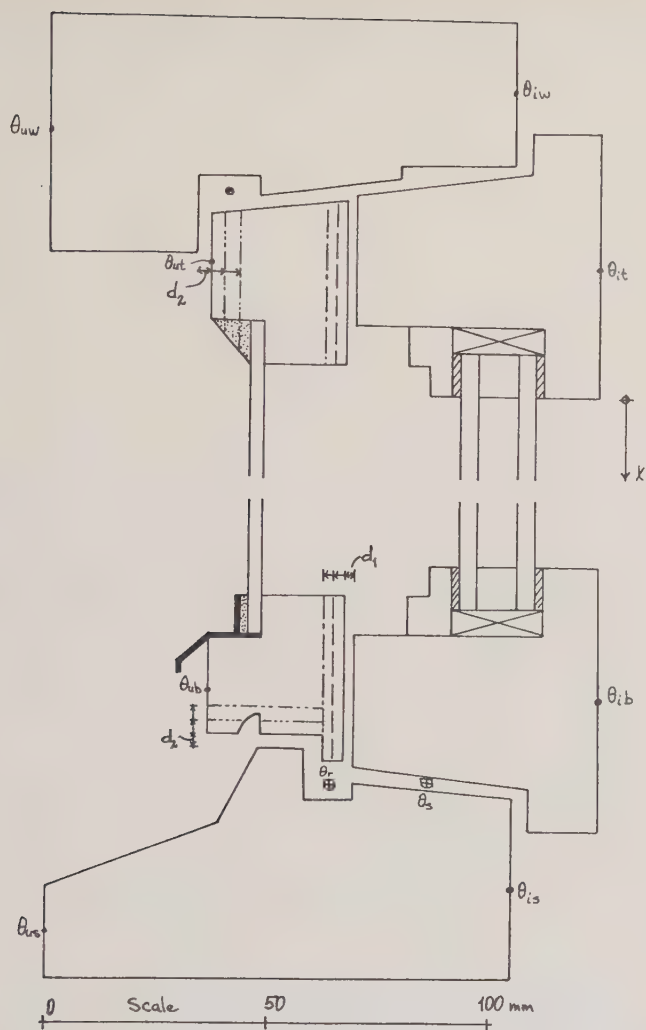


FIG.6.6.1 Section through 2+1 window. $d_1=2.4, 5.0$ or 7.0 mm (distance between casements), $d_2=2.8, 5.8$ or 9.4 mm (distance between outer casement and frame). $\oplus$ = temperature point (θ_d) in rainwater channel, $\boxplus$ = temperature point (θ_{space}) in gap between bottom rails of frame and inner casement.

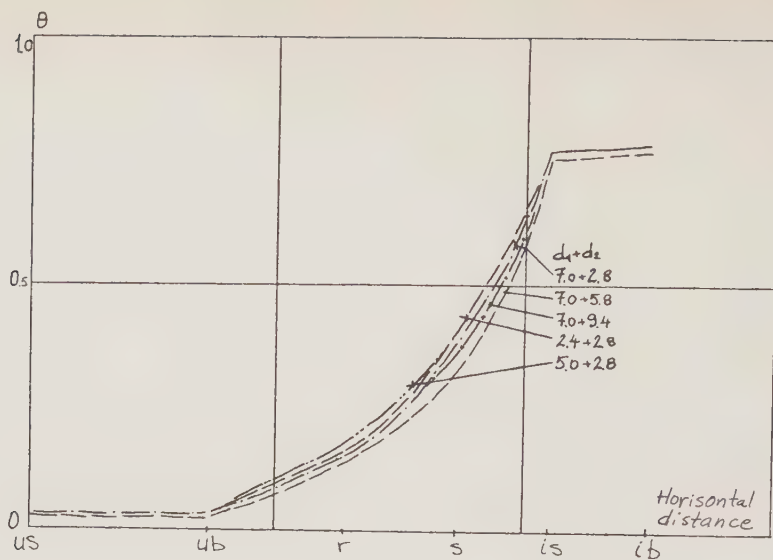


FIG.6.6.2 Temperature distribution in air space near the bottom rail. Variable air space width. Fan II, $\vartheta_U \sim -10^\circ\text{C}$.

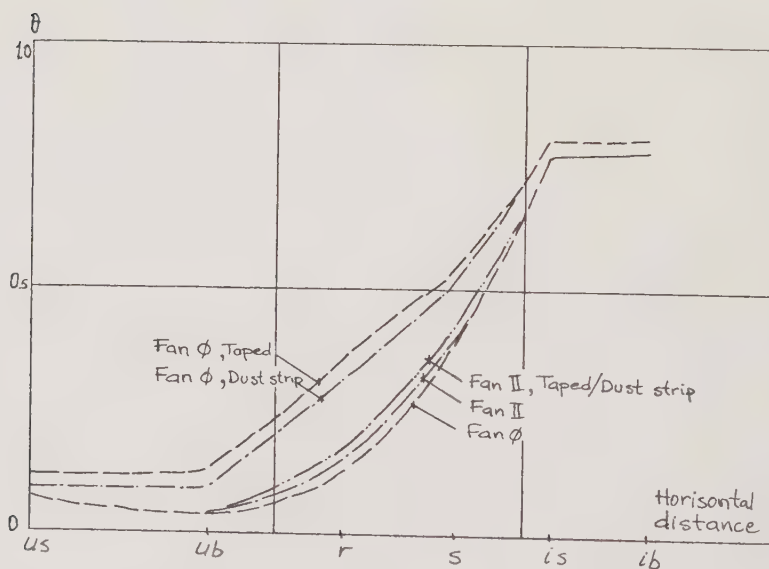


FIG.6.6.3 Temperature distribution in air space near the bottom rail. Effect of dust strip. $\vartheta_U \sim -10^\circ\text{C}$.

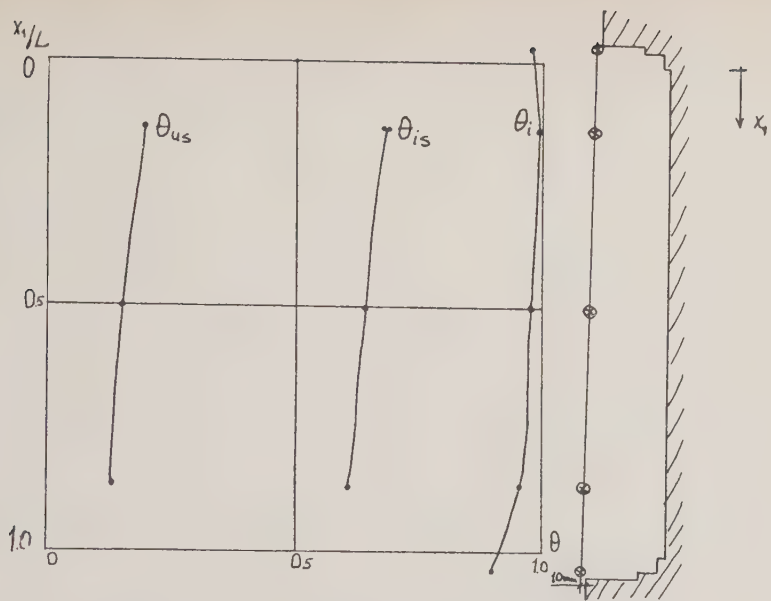


FIG.6.7.1 Temperature distribution along outside of glass, inside of glass and in the air 10 mm from the plane of the wall. L =height of air space.

$\theta = (\vartheta - \vartheta_U) / (\vartheta_i - \vartheta_U)$, where ϑ_U =outdoor air temperature 0.5 m from the facade, ϑ_i =temperature of indoor air perpendicular to window ($x_1/L=0.5$). Northerly window, coupled double glazed, no dust strip, mean distance between casements about 1.3 mm, air space ($L \times d \times b$) 1130x35x844 mm.

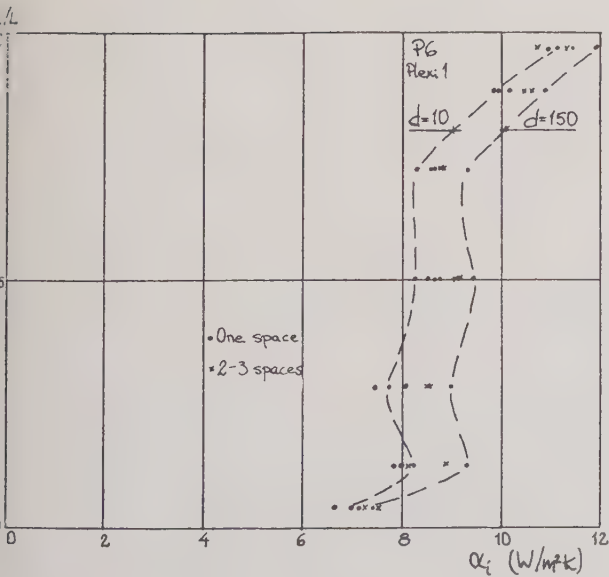


FIG.6.7.2 Internal surface coefficient of heat transfer (α_i) for test window P6 with air spaces varying between 10 and 150 mm.

7 DISCUSSION AND CONCLUSIONS

Calculation of convection in a space

The system of equations which can be set up for natural convection in a vertical space contains the Rayleigh number, the Prandtl number and the height/width ratio. When these nondimensional numbers and the boundary conditions are known, the equations can be solved. An iterative procedure must be applied and continued until the solution converges. The temperature distribution in the space and the heat flow along the surfaces can be calculated. The boundary conditions along the horizontal edges have been either adiabatic (insulated edges) or linear temperature distribution. In most cases, the vertical edges have assumed the nondimensional temperatures 0 and 1. In some cases the temperatures measured in the laboratory have been used for the vertical and horizontal edges.

In order that calculations may be put on a systematic basis, the flow has been classified in relation to the temperature gradient in horizontal direction in the centre of the space. Three different regimes can be distinguished, 1) the conduction regime in which heat transfer is mainly by conduction and the gradient is approximately -1 , 2) the transition regime with the gradient between -1 and 0 ; and 3) the boundary layer regime in which no heat transfer occurs in the central parts of the space, and the gradient is about 0 . As convection becomes more predominant (the Rayleigh number increases), flow changes in character from unicellular to multicellular. This secondary flow is formed at a critical value of the Rayleigh number. As the value of Ra further increases, weak tertiary cells are formed between the secondary cells. However, the formation of further cells depends to a large extent on the temperature distribution along the edges.

Calculations have been carried out with Rayleigh numbers between 1000 and 400000. The height/width ratio has been varied between 1 and 120. The Prandtl number has been varied between 0.69 and 0.89. Jonsson (1985) sets out all the calculated cases, while this presentation describes only a selection.

The way temperature distribution and heat flows change as convection increases can be studied in the calculations. The horizontal temperature gradient changes from -1 to 0 and the temperature along the vertical line in the centre of the space is no longer constant but varies with the height. By changing the height/width ratio, flow in the space can be altered and classified in a different flow regime.

The calculated cases and the classification into flow regimes given in the literature have been compared. The boundary between the conduction and transition regimes has been displaced laterally by a relatively large amount. The reason for this is that the classification is based on experimental investigations while this study comprises theoretically calculated values. A comparison of values measured in the laboratory and the above classification exhibits better agreement.

As Ra increases, the maximum vertical rate of flow increases and is displaced towards the vertical edges. The thickness of the boundary layer decreases.

Calculations show that under certain conditions flow becomes unstable and a multicellular convection pattern is formed. This occurs in particular for values of L/d around 20 and for values of the Rayleigh number between approximately 12000 and 40000 . If the height (L) is maintained constant and the width (d) is varied, heat flow at first diminishes as the width of the space increases in the conduction regime. When the width of the space is so large that the flow is located in the transition or boundary layer regime, heat flow slightly increases or remains relatively constant. Heat flow should be a minimum at the point where the conduction regime changes into the transition regime. Under certain conditions multicellular convection occurs here.

When convection commences at the bottom corner on the warm side (or the top corner on the cold side), the starting corners, air is rapidly heated (cooled) and heat flow at these points is considerable. It then decreases and is a minimum at the other corners, the departure corners. When a multicellular flow is formed, the curve of heat flow develops pronounced peaks. At the starting

corner the fluid is accelerated upwards (downwards) and the heat flow thus has a maximum value at the corner. The magnitude of this depends on the character of the flow, i.e. the Rayleigh number and the height/width ratio. Since the heat flow at the starting corner has a peak, subdivision of the space into a number of spaces gives rise to a number of peaks at the starting corner and the total heat flow will therefore be higher.

Heat flow due to convection can be expressed by means of the Nusselt number. When $Nu=1$, there is only conduction, and the greater the value of Nu becomes, the more pronounced is convection. The value of the Nusselt number averaged over the height can be written as

$$Nu_d = A Ra_d^B Pr^C (L/d)^D$$

The constants A , B , C and D have been assigned different values by different authors. The formulae given by Thomas (1970) agree quite well with measurements and calculations.

$$Nu_d = 0.59 (Ra_d)^{0.1} (L/d)^{-0.05} \quad \text{for the conduction regime according to FIG.3.2.1}$$

$$Nu_d = 0.20 (Ra_d)^{0.29} (L/d)^{-0.24} \quad \text{for the transition regime according to FIG.3.2.1}$$

$$Nu_d = 0.29 (Ra_d)^{0.26} (L/d)^{-0.24} \quad \text{for the boundary layer regime according to FIG.3.2.1}$$

The values of the Nusselt number according to the above appear to be about 5% too large in relation to measurements made in the laboratory.

If gases other than air are used in the space, conductive and convective heat transfer change. Properties such as thermal conductivity, density, viscosity and specific heat capacity vary a lot from gas to gas. The share due to conduction depends on thermal conductivity, while the share due to convection is a function of the Rayleigh number where $Ra = C_{fluid} \Delta T d^3$, and

C_{fluid} is a constant whose value depends on the gas used. On the other hand, the Prandtl number is relatively constant. For the gases studied, air, Ar, CO_2 , SO_2 and SF_6 , Pr varies between 0.69 and 0.89. The values of C_{fluid} for SF_6 is however about 35 times as high as for air. This means that for the same width, convection in a space containing SF_6 is more pronounced. The calculations suggest that for small space widths a gas of low thermal conductivity such as SF_6 should be chosen, while for greater widths a gas like Ar should be used.

For test windows P1-P4 calculations have been carried out with the temperature distribution along the edges, Ra and L/d selected on the basis of measurements in the laboratory. For all cases the character of flow is unicellular. If an attempt is made to fit the measured temperatures to a space with a fixed value of L/d (e.g. $L/d=10, 20$, etc) this easily gives rise to a multicellular pattern which in turn results in a higher value of Nu . It follows from this that if different boundary conditions give rise to approximately the same convection pattern, the average Nusselt number will be about the same. At high values of Rayleigh number, even calculations with the temperature 1 or 0 along the vertical edges may exhibit a multicellular pattern, which yields a higher value of the Nusselt number than the calculations relating to the test windows.

For a linear temperature distribution along the horizontal edges, the average Nusselt number will be smaller if some of the heat flow passes into the space at the bottom horizontal boundary surface and passes out of the space at the top horizontal surface. In this way the value of Nu for the inner vertical surface decreases.

The vertical distribution of the Nusselt number changes depending on the way temperature conditions along the edges are selected. At temperatures 1 and 0 along the vertical edges, a pronounced peak is formed at the bottom corner. If temperatures are selected for a test window, the calculated curve for the Nusselt number will have little variation along its central portion. A comparison of measured values of heat flows for the test windows P1-P4 and calculations based on measured temperatures along the edges

produces relatively poor agreement. It should be noted however that the calculated curve of the Nusselt number depends to a high degree on temperature distribution along the edges. The way this distribution line is selected depends very greatly on the way in which heat flow due to convection is distributed along the inside.

Calculations of flows along the edges

The frame+casement normally constitute a small proportion of the total surface of the window, and a change in the flow through the frame or casement will therefore only give rise to a small change in the thermal transmittance (k) for the window as a whole. Surface temperatures may however change so that there is a risk of condensation. In addition, improved insulation of the window must result in a reduction of heat flow through both frame+casement and through the glass. In calculating areas for k -values, the window has been projected on to a vertical surface. Air temperatures of 20°C and -20°C have been used.

The material selected for the spacer between the panes of glass exerts comparatively little influence on the thermal transmittance (k) of the window, but it gives rise to quite a large change in the flow through the frame and the minimum temperature. If the spacer is changed from a thermal insulation material to aluminium, the k -value of the window changes by about 3%, the k -value of the frame by 29%, and the minimum temperature drops by 5.4°C . For materials of relatively good thermal conductivity (aluminium, steel), it makes comparatively little difference whether the spacer is homogeneous, a hollow profile or a metal strip. Fusing together the ends of the panes of glass produces results midway between the extreme values.

Changing the distance between panes by means of a hollow aluminium profile as the spacer produces a large change in the thermal transmittance (k) of the window, a change from $d=6$ to $d=15$ mm causes a change of 11%. On the other hand, the minimum temperature is practically unchanged at about $+0.7^{\circ}\text{C}$ for two panes and about $+1.7^{\circ}\text{C}$ for three panes. In relation to the k -value

of the central portion of the window, the surface temperature rises by about 4°C on changing from two to three panes of glass. The minimum temperature however rises by only 1°C which is due to the fact that aluminium profiles are excellent conductors of heat. The width of glazing rebate has very little effect on the k-value of the window, while the minimum temperature is changed to some extent. Since this changes by just over 2°C when the glazing rebate is increased from 10 to 16 mm, this should be borne in mind when surface temperatures are measured on a sealed unit.

Whether the external glazing bead is wood or aluminium has little effect on a spacer with $\lambda=0.16\text{ W/mK}$. If the spacer is aluminium, the k-value increases by about 5% and the minimum temperature drops by 3°C as the glazing bead is changed from wood to aluminium. On the other hand, a metal weatherbar should not extend below the sealed unit as this causes the k-value of the window to increase (by about 6%) and the minimum temperature to drop (by 3.5°C).

The position of the sealed unit, in the centre or displaced towards the outside air, gives rise to only marginal changes for frames of wood, plastics, steel or aluminium. When the sealed unit is positioned on the outside, the minimum temperature is a little higher than when it is positioned in the centre. If an aluminium construction is to have a low thermal transmittance, thermal insulation material must be placed between the aluminium profiles. A steel profile is often inserted into plastics profiles for frames or casements to strengthen these. This increases the k-value by 4-5%.

On the basis of selected input data, the thermal transmittance (k) obtained for the window as a whole was 3.04, 3.11 and 3.33 $\text{W/m}^2\text{K}$ for frames of wood, plastics and aluminium respectively, i.e. a maximum change of 10%. The minimum temperatures were 3.4 , 3.5 and 5.8°C respectively, i.e. the aluminium frame which produces the highest k-value gives rise to the highest minimum temperature. The reason for this is that a large internal aluminium surface is capable of heating even the corner between the glass and the frame.

A comparison of two inward opening double glazed coupled windows, one comprising two wooden casements and the other with the outer casement of aluminium, shows that the aluminium casement gives rise to increased heat flow, and the k-value rises by 3%.

Three inward opening triple glazed wooden windows have been studied, one coupled window (without ventilation), one coupled window with a single glazed outer casement and a double glazed inner casement (2+1 window), and one with a triple glazed sealed unit. For the frame+casement, the k-values were 2.33, 2.35 and 2.79 $\text{W/m}^2\text{K}$ respectively. The reason for the high figure in the last case is the large heat flow next to the sealed unit. For the window as a whole the thermal transmittances (k) were 1.97, 2.22 and 2.54 $\text{W/m}^2\text{K}$ respectively. Some of this difference is due to differences in thermal resistance across the glazed portion. The k-value of the frame+casement is practically the same for the coupled window and the 2+1 window, while the window with the triple glazed sealed unit has a k-value about 19% higher and a lower surface temperature on the casement. An aluminium strip between the casement and frame raises the k-value of the frame+casement by about 0.3 $\text{W/m}^2\text{K}$, while the flow for the window as a whole increases by 5%. In addition the minimum temperature between frame and casement drops by about 5 °C.

For an outward opening 2+1 wooden window, the temperature profile is more uniform and the k-value lower (2.13 and 2.50 respectively) than in the case of an outward opening window with a triple glazed sealed unit. If the casement is clad with an aluminium construction, the difference in temperature and k-value is negligible. A comparison of outward and inward opening wooden windows shows that the k-values for the outward opening window are a little lower.

For an aluminium window with a triple glazed sealed unit the thermal transmittance (k) obtained was 2.72 $\text{W/m}^2\text{K}$, and that with a 2+1 construction was 2.61 $\text{W/m}^2\text{K}$. Two plastics windows with triple glazed sealed units had k-values of 2.12 and 2.73 $\text{W/m}^2\text{K}$. The difference is due to the way in which the gap between casement and frame is treated, and the insertion of a steel profile in one of the casements.

For a window of 1x1 m size, the k-value in these calculations has varied between 1.97 and 2.54 W/m²K (openable triple glazed wooden window), 2.76 and 2.84 W/m²K (openable double glazed wooden window), 2.61 and 2.72 W/m²K (openable triple glazed aluminium window) and 2.12 and 2.73 W/m²K (openable triple glazed plastics window). For fixed windows the k-values are about 3.0 and 2.2-2.3 W/m²K respectively for double and triple glazed windows.

Methods of measurement

Measurements by means of the guarded hot box provide a good overview of the entire window construction. Calibration of the hot box and the wall element is essential, and the measured results are in good agreement with the theoretically calculated values. For the test arrangement selected, the repeatability i.e. the accuracy in measuring the same object at different times is very good. The calculated mean thermal transmittance (k) varies within ± 0.05 W/m²K. Improvements can be made by controlling the temperature of the cold air and by varying the air flow rate on the cold side.

It must be borne in mind however that calibration sheets of different types and thicknesses exert an influence on the heat flow in the wall next to the window opening. It is important for this reason to establish a well defined procedure for calibration. When measurements are made on different types of window construction, the heat flow in the wall adjacent to the window frame will be changed. Because of this, the k-value of the window will contain an element due to the increased/decreased heat loss through the wall. This in itself raises no problem since a window in a building also affects heat transfer through the wall. What is important is to determine by how much the wall influences measurements when different types of tests are compared. When a detail such as the thermal resistance of the air space is studied it may be significant to make a correction with respect to the changed heat flow through the wall due to the alteration of the window construction.

Measurements of temperatures have been made with thermocouples of Type T, copper-constantan. If the different thermocouples are taken from the same roll, the readings will be very well grouped. In calibration tests the standard deviation has been less than 0.1°C . If greater precision is required, some types of calibration must be applied.

When surface temperatures are being measured, it is essential that the temperature sensors are in proper contact with the substrate. The best results have been obtained with sensors glued to the underlying material, taping on its own is not reliable. The special surface temperature sensors comprising thin thermocouples give better results than thermocouple wires. Sensors attached with silicon paste in order to improve contact between the sensor and the surface yield good results. One difficulty is keeping the silicon paste in place so that the sensor-surface contact is not changed. When surface temperatures of great accuracy were required, glued surface temperature sensors have been used.

Measurement of the heat flow rate at a point is very difficult. Two different methods have been used, measurement of the temperature difference across a sheet of known thermal conductivity, and measurement of the temperature with heat flow sensors according to the auxiliary wall principle. Both these methods have weaknesses and cannot be used indiscriminately. Measurement of the temperature difference across a sheet with a high degree of accuracy is difficult. The difference for plexiglass is little, about $2\text{--}4^{\circ}\text{C}$, and a small error in the surface temperature gives rise to the wrong value of the heat flow. This measurement provides the heat flow rate at a point.

In making comparisons in relation to a horizontal air space, it was found that the thermal resistance across a sheet of plexiglass was on average 16-19% less than the calculated value. The reason for this may be that the surface sensors are not exactly in the plane of the surface but protrude a little and thus measure too high a value of the temperature difference. This in turn give rise to an excessive heat flow rate.

When a heat flow sensor comprises a number of series of thermo-

couples, the output signal is stronger and the accuracy required is not as high. The measurement gives a mean value over a circle of approx 80 mm diameter. The sensor is however mounted on the outside of the surface which will create disturbances in convection and in the heat flow around the sensor. Use of a protective zone around the sensor increases heat flow by about 4%. In addition the output signal of the sensor changes depending on the thermal conductivity of the underlying material. The difference between an insulation material and glass may be as much as 4%. When measurements were made on materials of known thermal resistance, it was found that the measured thermal resistance was 11-23% too large. Measurements on a horizontal window construction also yield 16-22% higher values than theoretical calculations.

Measurements on test window P7 may be taken to illustrate the difficulties encountered in using heat flow sensors. The mean value of thermal resistance (m_{flm}) measured with three sensors placed along the vertical centre line is set out in APP. 6.4.1. The resistance has also been corrected by the factor 0.85-0.87 (see Subsection 6.7.1). In this way the following expression has been obtained for air spaces of a thermal resistance less than $0.2 \text{ m}^2\text{K/W}$

$$m_{sp} = 0.042 + 0.666 m_{flm} \quad R = 0.94$$

The mean values obtained are $m_{sp} = 0.155 \text{ m}^2\text{K/W}$ and $m_{flm} = 0.170 \text{ m}^2\text{K/W}$, i.e. a difference of about 10%.

For air spaces of thermal resistance between 0.28 and $0.52 \text{ m}^2\text{K/W}$ the expression is

$$m_{sp} = 0.073 + 0.682 m_{flm} \quad R = 0.996$$

with mean values $m_{sp} = 0.383 \text{ m}^2\text{K/W}$ and $m_{flm} = 0.455 \text{ m}^2\text{K/W}$ i.e. about 19% difference.

Since heat flow rate along the edges is greater than through the central portions, a resistance measured in the centre will be about 10% too high. If the thermal resistance of the central portions is made increasingly better while the resistance along the

edges remains the same, the percentage error in the resistance measured in the centre will be higher still. For a well insulated construction ($k=1.4 \text{ W/m}^2\text{K}$), there is a 25% difference ($m_{sp}=0.5$ and $m_{flm}=0.626$). This difference may be regarded as too large. A calculation based on the assumption that the edge influences 85% of the glass area gives rise to a difference of 10% and 13% for $k=2.8 \text{ W/m}^2\text{K}$ and $k=1.4 \text{ W/m}^2\text{K}$ respectively.

Experimental measurements

By constructing special test windows in which the heat flow rate is measured across the internal sheet of plexiglass the thermal properties of the window can be studied. As the width of air space is increased, the vertical temperature profile along the centre line of the air space changes from a constant value to an s-shaped curve and finally to a straight line where the temperature increases in proportion to the distance from the bottom edge. The temperature distribution in the horizontal direction through the centre of the window can be used to characterize flow into three regimes; the conduction, transition and boundary layer regimes. Measurements have shown that this division is a good way of characterizing flow.

By measuring the temperature difference across a sheet of plexiglass the heat flow rate can be calculated and variation of the flow in the vertical direction along different surfaces can also be determined. This method has been used primarily for a study of the general shapes of the curves and not for determination of the absolute levels. The latter requires extensive calibration. In the section concerning methods of measurement it has been pointed out that measurements of heat flow by temperature differences yield too high a value of the heat flow.

Factors which affect the heat flow curve are convection along both surfaces, radiant heat transfer to the surroundings, and heat transfer along the attachment of the glass (casement-frame). Convective circulation in the hot box is directed downwards along the window and is disturbed only at the junction between the top rail of the frame and the glass surface. This disturbance may

reduce heat flow slightly. Inside the air space in the corner heat flow is lower due to a reduction in radiant heat transfer (the spacer exerts a comparatively large influence) and a low value of the Nusselt number. When convection in the air space become stronger, heat flow in the bottom inner corner (starting corner) will increase. Heat flow rate then decreases as the distance from the bottom corner increases. As convection increases (e.g. as the width of air space increases), the curve has a greater slope.

A sheet placed inside the construction is influenced by convection currents on both sides. If these are about equal, the heat flow curve should be approximately symmetrical. It is evident from the measurements that heat flow along a sheet inside the construction is almost constant. This applies to two, three and four air spaces.

At the outermost glass, there is normally natural convection on one side of the glass and forced convection on the other. Owing to the use of fans, there is no stationary air on the cold side at the upper corner. The fans on the cold side give rise to an air current perpendicular to the window. If the fans are turned off the differences between heat flows decreases and the curve has a flatter slope. Since however the fans on the condenser in the cold room are still running, convection is to some extent forced. The case of a sheet inside the construction may be compared to natural convection on both sides.

With fans inside the hot box, circulatory motion and air velocity parallel to the window will increase. The shapes of the heat flow curves are about the same. It is at the upper edge that the rate of increase is not as large as in the rest of the curve. As has been pointed out before, heat flow is lower at the upper corner.

For test windows P1-P4 (FIG.6.2.12) the heat flow curves have a somewhat different appearance. Since test window P6 has two sheets of plexiglass against one in test window P2, the heat flow in P6 is about 10% less. The same difference is also found when P2 (FIG.6.2.12) and P6 (FIG.6.3.7) are compared.

Convection calculations using temperatures obtained in test windows P1-P4 yielded convection of mainly unicellular character in all cases. There are thus no air currents across the cell to give rise to local minima of the Nusselt number. It is however very difficult to ascertain by measurements where the local minima and the boundaries between cells are located, since the extent of a test window in the x direction is very small, only a few centimetres.

Naturally, the calculated values of α_i in FIG.6.7.2 have the same pattern as those in FIG.6.3.7-6.3.9, since the same value of heat flow through the inner glass has been used.

Tests have been made on hermetically sealed glass assemblies with different air space widths, different gases and low emissivity layers.

The minimum temperature varies between 0.5 and 0.6, and there is a tendency for a higher value of thermal resistance to give rise to higher minimum temperatures. The pronounced difference in temperature (4-7 °C) at the bottom rail of the frame is due to a combination of convection in the space, an aluminium spacer and a high surface coefficient of heat transfer in the corner. Although measurement of only the thermal resistance in the centre of a horizontal pane, with the heat transfer in the downward direction, eliminates these factors it produces a lower k-value which is not representative for the whole pane.

Changing convection in the hot box by fans to forced convection causes thermal resistance in a single space to decrease by about $0.01 \text{ m}^2\text{K/W}$.

The thermal resistance for the gases air and argon increases as the width of air space increases (6-15 mm), with argon having the highest resistance throughout. Sulphur hexafluoride has practically constant thermal resistance, and it is only for a space of 6 mm width that this gas has the highest thermal resistance. Thermal resistance may be said to increase by $0.02 \text{ m}^2\text{K/W}$ when air is replaced by argon, by $0.11 \text{ m}^2\text{K/W}$ if a pane is provided with a low emissivity layer, and by $0.20 \text{ m}^2\text{K/W}$ if both argon

and a low emissivity layer are used.

Comparison of calculated and measured thermal resistances

In Sections 6.3-6.4 the thermal resistances measured in the air spaces were compared with theoretical calculations using the expression

$$m_{\text{space}} = \frac{1}{\frac{\text{Nu} \lambda}{d} + 4 \cdot 5.67 \cdot 10^{-8} \epsilon_{\text{RES}} T_m^3}$$

where $\lambda = 0.0241 + 0.00008 \frac{t}{m}$ luft
 0.013 SF₆
 0.016 Ar

$$\epsilon_{\text{RES}} = 0.744 \quad (\epsilon_1 = \epsilon_2 = 0.853)$$

T_m = mean temperature over the space (K)

d = width of air space (m)

Nu = Nusselt number calculated according to a formula by Thomas (1970), see TAB.3.5.1.

d	Regime	Radi- ation W/m ² K	Ra	Nu	Cond+ conv W/m ² K	Thermal resistance		
						calc	meas	diff(%)
10	cond	3.28	2820	1.05	2.48	0.173	0.165	-5
20	bound	3.28	23200	1.57	1.85	0.194	0.174	-10
50	bound	3.28	374000	4.00	1.91	0.192	0.183	-5
100	bound	3.28	3.04 · 10 ⁶	8.10	1.94	0.191	0.188	-2
150	bound	3.28	10.2 · 10 ⁶	12.21	1.94	0.191	0.188	-2

It is thus particularly in the case of the air space of 20 mm width that agreement with calculations is not very good, agreement in the other cases is acceptable. The Nusselt number should have a value about 1.80 if the difference for $d=20$ mm is to be the same as in the other cases.

For the sealed units in Section 6.4 the figures are as follows:

Air	6	cond	3.80	235	1.0	4.11	0.126	0.109	-14
	12	cond	3.81	2240	1.03	2.12	0.168	0.148	-12
	15	cond	3.82	4550	1.12	1.84	0.176	0.159	-10
	12	cond	3.25	5480	1.13	2.25	0.181	0.158	-13
	15	trans	3.26	11000	1.21	1.90	0.193	0.168	-13
Ar	6	cond	3.21	870	1.00	2.65	0.170	0.148	-13
	12	trans	3.23	7800	1.22	1.61	0.206	0.187	-9
	15	trans	3.23	15200	1.40	1.48	0.212	0.187	-12
SF ₆	6	cond	3.21	25800	1.27	2.72	0.168	0.146	-13
	12	trans	3.21	$2.03 \cdot 10^5$	2.49	2.69	0.169	0.144	-15
	15	trans	3.23	$4.07 \cdot 10^5$	3.06	2.63	0.170	0.156	-8

The measured values are on average 11-12% lower than the calculated ones. This is largely due to the fact that the calculations take no account of the higher heat flow along the edges.

If the value $\epsilon_{\text{RES}}=0.15$ is applied for panes with low emissivity layers, the figures obtained are

D4-12,LE	cond	0.66	7240	1.16	2.30	0.336	0.262	-22
D4-12,LE,Ar	cond	0.66	9860	1.20	1.59	0.443	0.346	-22

and for two air spaces

T4-12	cond	3.56	3800	1.09	2.16	0.174		
	cond	3.00	3800	1.09	2.16	0.193		
						<u>0.367</u>	0.312	-15
T4-12,Ar	cond	3.58	4650	1.11	1.47	0.198		
	cond	2.98	4650	1.11	1.47	0.224		
						<u>0.422</u>	0.344	-17
T4-12,Ar,LE	cond	0.725	6400	1.15	1.52	0.443		
	cond	2.94	4500	1.11	1.47	0.226		
						<u>0.669</u>	0.520	-22

Panes which have better insulation in their central portions will, because of the practically constant heat flow along the edges, exhibit a greater difference between calculated and measured values. It is also evident that panes with low emissivity layers have an even greater difference; the value of ϵ_{RES} used should perhaps be a little higher. The emissivity of the low emissivity layer (kappa energi) was found at department of Technology, Uppsala University, to be 0.13.

Condensation

In order that the risk of condensation may be ascertained, it is necessary for the least surface temperature to be known. It is not sufficient to calculate surface temperature on the basis of the thermal transmittance (k) of the window and the internal surface resistance, since this yields an average surface temperature. A low surface temperature may be caused by a number of factors, such as convection in the air space, the shape of the junction with frame or casement, low surface coefficients of heat transfer in the corners, or insufficient insulation.

Owing to convection, the temperature at the bottom on the inside of the glass will drop, and as convection becomes more pronounced (the Rayleigh number increases), the temperature will drop further. When convection can be assigned to either the transition or boundary layer regime, minimum temperature remains approximately constant. Test window P6 may be taken as an example. In this the nondimensional minimum temperature (θ_{30}) is 0.64 for a space width of 10 mm and is then relatively constant, approximately 0.61, for 20–40 mm. This temperature (θ_{30}) is measured about 30 mm from the inner bottom corner. For the test windows there is a general tendency for the minimum temperature to rise as the width of air space (d) increases for $d > 40$ mm. In general, the minimum temperature is least for widths between 20 and 40 mm, which means that convection is in the transition regime. Generally speaking, the minimum temperature is a function of the k -value of the construction (mean surface temperature) and the effects at the boundary due to the design or convection (drop in temperature), i.e.

$$\theta_{\min} = \theta_{\text{mean}} - \Delta\theta_{\text{bound}}$$

There is thus a large rise in $\Delta\theta_{\text{bound}}$ when convection starts ($d=10$ to 20 mm), after this the change is less. Since θ_{mean} depends on the k -value of the construction, it increases most in the beginning ($d=10$ - 20 mm) and then becomes constant for $d=80$ - 150 mm.

The nondimensional minimum temperature for a certain air space width increases when the temperature difference across the window increases. This can be explained by the fact that thermal resistance in the air space increases as the mean temperature drops. Since the surface resistances remain approximately the same, this means that the mean surface temperature and also the minimum temperature rises as the mean temperature drops.

For constructions comprising a number of air spaces, it is the inner space which seems to be critical with regard to the minimum temperature. For two air spaces, the minimum temperature is $\theta_{30}=0.73$ and 0.71 when the width of the inner air space is $d=10$ and $d=20, 40$ mm respectively. For three spaces the temperature is $\theta_{30}=0.80$ and 0.77 for inner space widths of $d=10$ and 20 mm respectively. In the case of four spaces, the temperatures for the same space widths are $\theta_{30}=0.83$ and 0.80 respectively.

For hermetically sealed glass assemblies, an aluminium spacer causes the minimum temperature to drop further. There is a general tendency for the minimum temperature to rise as the k -value decreases. It was found by correlation analysis that the values in TAB.6.4.1 can be expressed by

$$\theta_{30}=0.87-0.092 k$$

$$\theta_0=0.73-0.07 k$$

where θ_0, θ_{30} = nondimensional temperatures at distances of 0 and 30 mm respectively from the bottom corner

k = the mean k -value of the window
 $(m_i+m_u=0.17 \text{ m}^2\text{K/W})$

The corresponding values for test windows P1-P6 are

$$\theta_{30}=0.90-0.113 \text{ k}$$

If the surface temperature is calculated using the mean k-value, we have ($m_1=0.115 \text{ m}^2\text{K/W}$)

$$\theta_{\text{mean}} = 1-0.115 \text{ k}$$

When test windows P1-P6 and P7 are compared, it is seen that the aluminium spacer causes the temperature θ_{30} to drop by 0.02-0.06.

It is also to be noted that a lower k-value has greater effect on θ_{mean} compared to θ_{30} or θ_0 .

If only the expression for θ_{mean} is used for calculation of θ_{30} or θ_0 , a local k-value must be used (k_{30}). We thus have for $\theta_{30}=\theta_{\text{mean}}$ that

$$\begin{aligned} k_{30} &= 0.87+k && \text{for test windows P1-P6} \\ k_{30} &= 1.13+0.8k && \text{for test window P7} \end{aligned}$$

This is equivalent to a resistance (m_{30}) for the construction equal to

k	P1-P6 m_{30}	P7 m_{30}	m_{mean}
1.5	0.25	0.26	0.50
2	0.18	0.20	0.38
3	0.09	0.11	0.16

If the same calculation is carried out for $\theta_0=\theta_{\text{mean}}$ for test window P7, the resistance of the construction is found to be 0.14, 0.11 and 0.07 respectively for $k=1.5$, 2.0 and 3.0 $\text{W/m}^2\text{K}$. This is approximately equal to the resistance through the spacer.

The material used in the casement or frame and in the spacer between the panes is of great significance. If the wooden frame is replaced by one of expanded polyurethane, the minimum temperature

(θ_{30}) decreases from about 0.60 to 0.58. The influence of the spacer and the material in the frame+casement can also be studied in TAB.4.3.1-4.3.2. For a frame and spacer of wood, the minimum temperature (case 1-3) is $\theta_0=0.61$ which is about the same as the measured values. For a spacer of aluminium the calculations (case 3-3 in TAB.4.3.2) yield $\theta_0=0.47$ while measurements (case D4-12 in APP.6.4.1) give $\theta_0=0.48$.

Width of space between coupled casements

It must be pointed out to start with that, apart from the width of the space between the casements, the ventilation between coupled casements also depends on wind speed and direction and the design of the window construction. This investigation has been carried out on an inward opening 2+1 window (double glazed inner casement and single glazed outer casement) in a hot box.

The outside temperature has very little effect on the thermal transmittance (k). Wind speed has a comparatively small influence, a rise from 0 to 1 m/s changes the k -value by about 3%. If the width of space between the casements is increased from 2.4 mm to 7 mm, the k -value changes by only 3%, and if the gap between the outer casement and the frame is then altered from 2.8 mm to 9.4 mm, the k -value increases by a further 3%. The influence of a dust strip (for space widths of 2.4-7.0 mm) at these wind speeds is comparable to that due to sealing the space. A minimum pressure across the dust strip is presumably needed in order that an air current through this should be set up. If this is so, it is likely that ventilation will function better at high wind speeds. The amount by which sealing of the air space or application of a weatherstrip reduces the k -value depends on wind speed. At a speed of 1 m/s the k -value is reduced by about 3%, and at a wind speed of 0 m/s the reduction is about 7%. When there is some force in existence, either due to the wind or to natural convection between the panes (chimney effect), cold air will circulate past and between the bottom rails of the frame and casement. When the air space is sealed off and there is no wind, this circulation ceases, the temperatures at the bottom rails increase and heat flow will therefore decrease.

Surface coefficients of heat transfer

Measurements on windows in a room (Section 6.7) showed that the inner surface coefficient of heat transfer is relatively constant over the surface. At the upper and lower edges of the windows, there was some variation in α_i in the horizontal direction. At midheight α_i was practically constant. On average, the values obtained along the vertical centre line were $\alpha_i=9.0, 8.0, 7.75$ and $8.25 \text{ W/m}^2\text{K}$ for the top rail, centre, bottom rail and the window as a whole respectively. This is however a small variation.

Measurements on sealed units (test window P7) in a hot box yielded values of α_i between 7.95 and $8.45 \text{ W/m}^2\text{K}$, i.e. $m_i=0.126\text{--}0.119 \text{ m}^2\text{K/W}$. The outer surface coefficient of heat transfer α_u varied between 21.6 and $24.0 \text{ W/m}^2\text{K}$, i.e. $m_u=0.046\text{--}0.042 \text{ m}^2\text{K/W}$. This means that m_i+m_u is relatively constant at a level of $0.172\text{--}0.161 \text{ m}^2\text{K/W}$.

For the other test windows, P1-P5, the value of m_i+m_u is between 0.155 and $0.175 \text{ m}^2\text{K/W}$. Test window P6 is slightly higher as $m_i+m_u=0.17\text{--}0.18 \text{ m}^2\text{K/W}$ for one air space. If the number of spaces is increased, there is a tendency for m_i+m_u also to increase, for four air spaces the value of $m_i+m_u=0.20\text{--}0.22 \text{ m}^2\text{K/W}$.

If a value of α_i is calculated (see Section 4.2), for a difference in air and surface temperatures of $10\text{--}16^\circ\text{C}$ the value of α_i is $8.2\text{--}8.5 \text{ W/m}^2\text{K}$ ($m_i=0.122\text{--}0.118 \text{ m}^2\text{K/W}$). In selecting values of α_i and α_u for calculations of heat flows near the casement and frame (Section 4.3), account has been taken of both the above theoretical values and the experimental values of α_i and α_u .

During laboratory measurements, distribution of α_i in the vertical direction may be said to vary between 9.7 and $7.8 \text{ W/m}^2\text{K}$. These figures include a correction of 1.15 (see the section on methods of measurement). The outer surface coefficient of heat transfer α_u varies in the cold room between about 14 and $20 \text{ W/m}^2\text{K}$. If the fans in the cold room are turned off, the amplitude

of the α_u curve is reduced by about 60%, and the mean value is about the same as on the warm side.

For the horizontal part of the frame or casement the inner surface coefficient of heat transfer is a function of the material selected and the construction. If the window has one casement and frame of wood or plastics, α_i may vary between 2.13 and 3.13 $\text{W/m}^2\text{K}$ ($m_i=0.47-0.32 \text{ m}^2\text{K/W}$) for a double glazed construction and between 2.86 and 4.00 $\text{W/m}^2\text{K}$ ($m_i=0.35-0.25 \text{ m}^2\text{K/W}$) for a triple glazed construction. The mean values are 2.50 $\text{W/m}^2\text{K}$ ($m_i=0.40 \text{ m}^2\text{K/W}$) and 3.45 $\text{W/m}^2\text{K}$ ($m_i=0.29 \text{ m}^2\text{K/W}$) respectively. If the casement and frame are aluminium, the values of $\alpha_i=3.33 \text{ W/m}^2\text{K}$ and 3.85 $\text{W/m}^2\text{K}$ respectively for double and triple glazed windows. If there is no break in the aluminium profiles in frame and casement between the warm and cold sides (uninsulated construction), the value of α_i is raised to about 4.55 $\text{W/m}^2\text{K}$.

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APPENDIX

Tabulated data for test windows P1, P2, P3, P4, P5, P6, P7 and the coupled windows in sections 6.6 and 6.7.1.

Case	$\Delta\dot{V}$	θ_{us}	θ_{is}	β_h	β_v	Plexi1	Plexi1	Hot-box	Hot-box	Hot-box	k	Min.			
				sp1	sp1	m _{sp1}	spec flow	m ₁ +m _u	m _{sp1}	spec flow	m ₁ +m _u = 0.2	temp. x= 0.973			
P6+L10+G3	47.7	0.105	0.687	-	-0.56	0.16	0.165	-	2.963	0.153	0.178	-	2.732	2.421	0.632
P6+L10+G3	37.9	0.104	0.677	-	-0.62	-0.08	0.165	-	2.916	0.155	0.174	-	2.755	2.449	0.631
P6+L10+G3	28.3	0.114	0.663	-	-0.66	-0.42	0.156	-	2.931	0.164	0.165	-	2.747	2.503	0.629
P6+L20+G3	47.9	0.111	0.686	-	-0.55	-0.07	0.158	-	3.024	0.158	0.178	-	2.688	2.412	0.612
P6+L20+G3	38.2	0.113	0.680	-	-0.59	-0.08	0.160	-	2.959	0.164	0.178	-	2.646	2.414	0.607
P6+L20+G3	27.5	0.104	0.668	-	-0.58	-0.16	0.161	-	2.919	0.159	0.171	-	2.747	2.469	0.593
P6+L20+G3	20.2	0.097	0.658	-	-0.66	-0.44	0.167	-	2.821	0.163	0.175	-	2.688	2.444	0.587
P6+L30+G3	48.0	0.105	0.684	-	-0.47	-0.26	0.158	-	3.047	0.158	0.180	-	2.667	2.401	0.608
P6+L30+G3	38.5	0.108	0.677	-	-0.51	-0.29	0.158	-	3.002	0.163	0.179	-	2.646	2.410	0.602
P6+L30+G3	27.8	0.106	0.666	-	-0.56	-0.31	0.159	-	2.932	0.167	0.176	-	2.639	2.427	0.591
P6+L30+G3	20.2	0.109	0.664	-	-0.57	-0.38	0.166	-	2.806	0.170	0.177	-	2.618	2.431	0.589
P6+L40+G3	47.5	0.109	0.704	-	-0.35	0.16	0.172	-	2.919	0.154	0.191	-	2.625	2.343	0.626
P6+L40+G3	38.3	0.105	0.701	-	-0.38	0.13	0.169	-	2.966	0.155	0.192	-	2.604	2.334	0.626
P6+L40+G3	28.0	0.107	0.692	-	-0.39	0.18	0.169	-	2.914	0.155	0.184	-	2.674	2.387	0.616

APP.6.2.1

Test window P1, air space 1125x1125 mm Indoor air $\theta_{i1}=1$, outdoor air $\theta_u=-0. \theta_{is}$ and θ_{us} are surface temperatures along the warm and cold surface. The horizontal and vertical gradients are calculated in the centre of the air space. P=plexiglass, G=glass and L=air space. $\Delta\dot{V}$ =difference between indoor-outdoor air temperature ($^{\circ}\text{C}$). Plexi 1 refers to air space. Hot box refers to mean power through air space+spacer (1152x1150 mm).

Case	$\Delta \bar{V}$	θ_{us}	θ_{is}	θ_{g2}	β_h sp1	β_v sp1	Plexi 1 m _{sp1}	Plexi 1 m _{sp2}	Plexi 1 spec flow	Hot-box m _i +m _u	Hot-box m _{sp1}	Hot-box m _{sp2}	Hot-box spec flow	k m _i +m _u = 0.2	Min. temp.x= 0.964
P6+L10+G4	48.1	0.119	0.696	-	-0.47	-0.28	0.156	-	2.913	0.155	0.177	-	2.725	2.429	0.622
P6+L10+G4	38.5	0.116	0.689	-	-0.50	-0.28	0.166	-	2.888	0.157	0.176	-	2.725	2.434	0.622
P6+L10+G4	28.1	0.114	0.672	-	-0.53	-0.16	0.159	-	2.904	0.159	0.167	-	2.778	2.494	0.613
P6+L20+G4	48.4	0.112	0.690	-	-0.38	0.10	0.158	-	3.036	0.157	0.179	-	2.688	2.409	0.601
P6+L20+G4	38.9	0.110	0.683	-	-0.40	0.06	0.160	-	2.979	0.159	0.178	-	2.681	2.414	0.597
P6+L20+G4	28.0	0.114	0.673	-	-0.44	0.00	0.157	-	2.945	0.167	0.176	-	2.639	2.430	0.586
P6+L30+G4	47.6	0.115	0.690	-	-0.37	0.46	0.155	-	3.062	0.163	0.183	-	2.604	2.376	0.600
P6+L30+G4	28.1	0.118	0.675	-	-0.45	0.47	0.158	-	2.918	0.159	0.166	-	2.786	2.501	0.586
P6+L40+G4	37.9	0.110	0.686	-	-0.35	0.54	0.159	-	2.996	0.163	0.184	-	2.604	2.375	0.596
P6+L40+G4	27.9	0.112	0.675	-	-0.40	0.52	0.159	-	2.937	0.162	0.174	-	2.695	2.441	0.585
P6+L50+G4	48.4	0.116	0.698	-	-0.18	0.76	0.164	-	2.953	0.163	0.190	-	2.564	2.344	0.606
P6+L50+G4	28.5	0.111	0.680	-	-0.29	0.72	0.165	-	2.885	0.167	0.185	-	2.577	2.375	0.592
P6+L20+G5+L20+G4	38.8	0.070	0.785	0.443	-0.20	0.66	0.146	0.190	1.911	0.167	0.165	0.214	1.704	1.613	0.694

APP.6.2.2 Test window P2, air space 841x1123 mm. Indoor air $\theta_{i1}=1$, outdoor air $\theta_{u}=0$. θ_{is} and θ_{us} are surface temperatures along the warm and cold surface. The horizontal and vertical gradients are calculated in the centre of the air space. P=plexiglass, G=glass and L=air space. $\Delta \bar{V}$ =difference between indoor-outdoor air temperature ($^{\circ}\text{C}$). Plexi 1 refers to air space. Hot box refers to mean power through air space+spacer (866x1150 mm).

Case	$\Delta \bar{V}$	θ_{us}	θ_{is}	θ_{g2}	$\beta_{h, sp1}$	$\beta_{v, sp1}$	Plexi1 m_{sp1}	Plexi1 spec flow	Hot-box $m_{sp1} + n_U$	Hot-box m_{sp2}	Hot-box spec flow	k $m_{sp1} + m_{sp2} = 0.2$	Min. temp. $x = 0.946$
P6+L10+G4	48.4	0.103	0.685	-	-0.53	0.60	0.163	-	0.153	0.178	-	2.420	0.611
P6+L10+G4	38.7	0.100	0.680	-	-0.59	0.48	0.163	-	0.155	0.179	-	2.414	0.609
P6+L10+G4	28.1	0.100	0.669	-	-0.59	0.31	0.156	-	0.158	0.172	-	2.455	0.608
P6+L20+G4	48.0	0.108	0.688	-	-0.49	0.51	0.158	-	0.165	0.189	-	2.341	0.602
P6+L20+G4	38.0	0.106	0.681	-	-0.53	0.47	0.157	-	0.166	0.187	-	2.351	0.596
P6+L30+G4	28.5	0.102	0.672	-	-0.46	0.62	0.154	-	0.164	0.180	-	2.395	0.588
P6+L40+G4	48.2	0.102	0.687	-	-0.03	0.67	0.161	-	0.165	0.194	-	2.313	0.593
P6+L40+G4	28.6	0.106	0.676	-	-0.26	0.64	0.155	-	0.163	0.179	-	2.404	0.583
P6+L50+G4	47.8	0.106	0.691	-	0.00	0.67	0.162	-	0.163	0.193	-	2.322	0.598
P6+L50+G4	38.6	0.103	0.683	-	-0.01	0.63	0.159	-	0.169	0.195	-	2.307	0.591
P6+L20+G4+L20+G4	48.8	0.059	0.791	0.412	-0.29	0.88	0.154	0.169	0.167	0.196	0.214	1.608	1.527
P6+L20+G4+L20+G4	39.2	0.058	0.785	0.406	-0.31	0.81	0.153	0.165	0.168	0.194	0.210	1.621	1.540
P6+L20+G4+L20+G4	28.5	0.060	0.780	0.399	-0.34	0.79	0.158	0.165	0.179	0.204	0.213	1.558	1.510

APP.6.2.4 Test window P3, air space 561x1123 mm. Indoor air $\theta_{i1}=1$, outdoor air $\theta_{u0}=0$. θ_{is} and θ_{us} are surface temperatures along the warm and cold surface. The horizontal and vertical gradients are calculated in the centre of the air space. P=plexiglass, G=glass and L=air space. $\Delta \bar{V}$ =difference between indoor-outdoor air temperature ($^{\circ}\text{C}$). Plexi 1 refers to air space. Hot box refers to mean power through air space+spacer (585x1150 mm).

Case	ΔT	θ_{us}	θ_{is}	θ_{g2}	β_h sp1	A_v sp1	Plexi1 m_{sp1}	Plexi1 m_{sp2}	Plexi1 spec flow	Hot-box m_i+m_u	Hot-box m_{sp1}	Hot-box m_{sp2}	Hot-box spec flow	k $m_i+m_u=0.2$	min. temp. 0.857
P6+L10+G4	49.0	0.113	0.693	-	-0.71	0.35	0.161	-	(3.067) 2.966	0.159	0.183	-	2.646	2.386	0.641
P6+L10+G4	39.0	0.110	0.683	-	-0.70	0.24	0.154	-	(3.076) 3.066	0.160	0.178	-	2.667	2.412	0.641
P6+L10+G4	28.7	0.115	0.673	-	-0.68	0.12	0.145	-	(3.098) 3.153	0.165	0.171	-	2.681	2.446	0.640
P6+L20+G4	38.5	0.111	0.689	-	-0.46	0.36	0.152	-	(3.127) 3.126	0.173	0.196	-	2.445	2.291	0.620
P6+L20+G4	28.9	0.117	0.680	-	-0.52	0.30	0.150	-	(3.134) 3.094	0.176	0.189	-	2.475	2.336	0.615
P6+L30+G4	39.4	0.116	0.685	-	-0.13	0.48	0.154	-	(3.142) 3.056	0.189	0.208	-	2.278	2.223	0.622
P6+L40+G4	38.6	0.109	0.692	-	-0.04	0.47	0.154	-	(3.096) 3.119	0.183	0.213	-	2.283	2.197	0.630
P6+L50+G4	38.7	0.128	0.694	-	-0.01	0.52	0.148	-	(3.097) 3.136	0.195	0.210	-	2.227	2.203	0.632
P6+L20+G4+L20+G4	47.8	0.081	0.788	0.440	-0.20	0.73	0.138	0.170	(2.162) 2.055	0.19	0.183	0.227	1.546	1.521	0.727
P6+L20+G4+L20+G4	38.8	0.075	0.785	0.435	-0.26	0.59	0.131	0.162	(2.148) 2.148	0.212	0.208	0.257	1.370	1.393	0.724

APP.6.2.5 Test window P4, air space 272x1123 mm. Indoor air $\theta_i=1$, outdoor air $\theta_u=0$. θ_{is} and θ_{us} are surface temperatures along the warm and cold surface. The horizontal and vertical gradients are calculated in the centre of the air space. P=plexiglass, G=glass and L=air space. ΔT =difference between indoor-outdoor air temperature ($^{\circ}\text{C}$). Plexi 1 refers to air space. Hot box refers to mean power through air space+spacer (300x1150 mm). The flow rate (plexi 1) calculated for $z=0.625$ and ($z=0.877$).

x_1/L	P1-10 d=10 mm		P1-20 d=20 mm		P1-30 d=30 mm		P1-40 d=40 mm		P1-50 d=50 mm	
	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}
0.0	0.159	1.148	0.229	1.292	0.240	1.306	0.200	1.289	0.189	1.299
0.034	0.126	1.121	0.174	1.232	0.186	1.251	0.161	1.241	0.154	1.251
0.096	0.066	1.072	0.075	1.123	0.087	1.151	0.090	1.154	0.090	1.162
0.216	0.007	1.019	0.002	1.026	0.010	1.037	0.021	1.057	0.025	1.073
0.377	-0.057	0.987	-0.055	0.968	-0.058	0.968	-0.048	0.972	-0.046	0.977
0.630	0.003	1.003	0.011	0.989	0.002	0.984	0.000	0.975	0.000	0.970
0.791	-0.024	0.989	-0.029	0.992	-0.030	0.979	-0.037	0.964	-0.042	0.947
0.911	-0.009	0.930	-0.029	0.916	-0.033	0.882	-0.043	0.873	-0.041	0.859
0.973	0.007	0.874	-0.043	0.795	-0.045	0.799	-0.048	0.812	-0.048	0.800
1.0	0.014	0.849	-0.049	0.742	-0.050	0.763	-0.051	0.730	-0.051	0.773
$\bar{\theta}$	-10.21	8.45	-10.10	8.45	-10.42	8.28	-10.48	9.20	-10.89	8.22
Ra	2890		22610		77500		190520		365990	
L/d	112.4		56.2		37.5		28.1		22.5	

APP.6.2.6 Nondimensional surface temperature θ in the air space for test window P1 with $L=1125$ mm and $d=10, 20, 30, 40$ and 50 mm. $\theta = (\bar{\theta} - \theta_{sp-}) / (\theta_{sp+} - \theta_{sp-})$.

x_1/L	P2-10 d=10 mm		P2-20 d=20 mm		P2-30 d=30 mm		P2-40 d=40 mm		P2-50 d=50 mm	
	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}
0	0.194	1.162	0.229	1.259	0.230	1.266	0.213	1.253	0.218	1.253
0.046	0.139	1.131	0.168	1.207	0.173	1.221	0.162	1.213	0.167	1.215
0.128	0.042	1.076	0.061	1.114	0.073	1.142	0.072	1.141	0.078	1.147
0.289	-0.014	1.006	-0.007	1.008	-0.001	1.013	0.006	1.042	0.007	1.061
0.505	-0.014	1.014	-0.007	1.006	-0.009	1.002	-0.005	1.005	-0.009	0.996
0.721	-0.011	0.995	-0.026	0.980	-0.028	0.968	-0.035	0.951	-0.040	0.943
0.881	-0.037	0.921	-0.058	0.895	-0.069	0.879	-0.071	0.862	-0.065	0.857
0.964	-0.077	0.815	-0.111	0.763	-0.119	0.756	-0.114	0.755	-0.114	0.751
1.0	-0.094	0.770	-0.134	0.706	-0.141	0.703	-0.133	0.708	-0.135	0.705
$\bar{\theta}$	-10.42	8.55	-10.84	8.24	-18.38	4.93	-10.08	8.60	-19.12	5.08
Ra	2870		22900		102620		177350		493110	
L/d	84.1		42.0		28.0		21.0		16.8	

APP.6.2.7 Nondimensional surface temperature θ in the air space for test window P2 with $L=841$ mm and $d=10, 20, 30, 40$ and 50 mm. $\theta = (\bar{\theta} - \theta_{sp-}) / (\theta_{sp+} - \theta_{sp-})$.

x_1/L	P3-10 d=10 mm		P3-20 d=20 mm		P3-30 d=30 mm		P3-40 d=40 mm		P3-50 d=50 mm	
	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}
0	0.178	1.172	0.216	1.262	0.201	1.261	0.196	1.246	0.183	1.233
0.071	0.117	1.133	0.144	1.201	0.139	1.209	0.137	1.198	0.130	1.188
0.196	0.007	1.065	0.018	1.092	0.028	1.118	0.033	1.114	0.035	1.108
0.438	-0.033	1.003	-0.034	1.002	-0.032	1.003	-0.024	1.022	-0.019	1.022
0.580	-0.005	1.003	-0.010	0.990	-0.015	0.979	-0.016	0.984	-0.014	0.989
0.821	-0.029	0.941	-0.039	0.907	-0.045	0.889	-0.055	0.882	-0.057	0.887
0.946	-0.052	0.827	-0.076	0.780	-0.073	0.774	-0.073	0.766	-0.075	0.772
1.0	-0.061	0.778	-0.092	0.726	-0.085	0.724	-0.080	0.716	-0.083	0.722
$\bar{\theta}$	-11.26	7.96	-10.30	8.33	-19.28	4.40	-19.61	4.50	-11.02	8.12
Ra	2980		22880		108310		263530		369910	
L/d	56.0		28.0		18.7		14.0		11.2	

APP.6.2.8 Nondimensional surface temperature θ in the air space for test window P3 with $L=561$ mm and $d=10, 20, 30, 40$ and 50 mm. $\theta = (\bar{\theta} - \bar{\theta}_{sp-}) / (\bar{\theta}_{sp+} - \bar{\theta}_{sp-})$.

x_1/L	P4-10 d=10 mm		P4-20 d=20 mm		P4-30 d=30 mm		P4-40 d=40 mm		P4-50 d=50 mm	
	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}	θ_{sp-}	θ_{sp+}
0	0.044	1.101	0.079	1.195	0.070	1.183	0.060	1.178	0.062	1.186
0.110	0.030	1.082	0.058	1.155	0.052	1.145	0.044	1.141	0.048	1.146
0.604	-0.031	0.995	-0.033	0.975	-0.032	0.975	-0.027	0.972	-0.015	0.968
0.857	-0.005	0.907	-0.036	0.841	-0.029	0.853	-0.025	0.862	-0.041	0.859
1.0	-0.010	0.858	-0.032	0.766	-0.028	0.784	-0.024	0.800	-0.056	0.798
$\bar{\theta}$	-10.85	8.13	-10.43	8.41	-10.83	8.18	-10.64	8.49	-9.92	8.58
Ra	2900		22590		77210		181940		335720	
L/d	27.3		13.6		9.1		6.8		5.5	

APP.6.2.9 Nondimensional surface temperature θ in the air space for test window P4 with $L=272$ mm and $d=10, 20, 30, 40$ and 50 mm. $\theta = (\bar{\theta} - \bar{\theta}_{sp-}) / (\bar{\theta}_{sp+} - \bar{\theta}_{sp-})$.

x_1/L	P1-20/20		L/d=56.2		x_1/L	P2-20/20		L/d=42.0	
	θ_{sp2-}	θ_{sp2+}	θ_{sp1-}	θ_{sp1+}		θ_{sp2-}	θ_{sp2+}	θ_{sp1-}	θ_{sp1+}
0.0	0.204	1.580	0.624	1.299	0.0	0.242	1.512	0.658	1.336
0.034	0.157	1.479	0.514	1.256	0.046	0.176	0.401	0.515	1.282
0.096	0.072	1.253	0.272	1.178	0.128	0.059	1.201	0.259	1.187
0.216	0.007	0.991	-0.016	1.060	0.289	-0.010	1.061	0.079	1.062
0.377	-0.036	0.870	-0.139	0.987	0.505	-0.008	0.957	-0.055	1.039
0.630	0.008	0.987	-0.014	1.004	0.721	-0.027	0.903	-0.124	0.945
0.791	-0.037	0.970	-0.032	0.976	0.881	-0.061	0.832	-0.216	0.800
0.911	-0.047	0.955	-0.049	0.827	0.964	-0.100	0.688	-0.401	0.586
0.973	-0.059	0.795	-0.220	0.647	1.0	-0.117	0.625	-0.482	0.493
1.0	-0.064	0.726	-0.294	0.569					
$\bar{\theta}$	-12.72	1.01	1.01	13.79		-12.29	2.11	2.11	13.32
Ra	16110		14930			17570		12440	

x_1/L	P3-20/20		L/d=28.0		x_1/L	P4-20/20		L/d=13.6	
	θ_{sp2-}	θ_{sp2+}	θ_{sp1-}	θ_{sp1+}		θ_{sp2-}	θ_{sp2+}	θ_{sp1-}	θ_{sp1+}
0.0	0.200	1.506	0.541	1.316	0.0	0.097	1.301	0.367	1.285
0.071	0.122	1.387	0.413	1.257	0.110	0.072	1.238	0.292	1.229
0.196	-0.016	1.177	0.189	1.153	0.604	-0.038	0.961	-0.048	0.974
0.438	-0.017	0.988	-0.013	1.033	0.857	-0.047	0.756	-0.298	0.755
0.580	-0.019	0.913	-0.093	1.001	1.0	-0.052	0.641	-0.439	0.632
0.821	-0.019	0.841	-0.170	0.843					
0.946	-0.048	0.661	-0.362	0.655					
1.0	-0.061	0.584	-0.445	0.575					
$\bar{\theta}$	-13.4	0.38	0.38	13.03		-11.95	1.89	1.89	13.21
Ra	17480		13610			17520		11840	

APP.6.2.10 Nondimensional surface temperature θ in the air space for test windows P1, P2, P3 and P4 with $L=1125, 841, 561$ and 272 mm respectively and $d=20+20$ mm.

$$\theta = (\bar{\theta} - \bar{\theta}_{sp-}) / (\bar{\theta}_{sp+} - \bar{\theta}_{sp-}).$$

Case	ΔT	θ_{us}	θ_{is}	β_h	m_{sp1} plexi 1	m_{sp1} plexi 2	Spec flow plexi 1	Spec flow plexi 2	Hot-box m_{i+u}	Hot-box m_{sp1}	Hot-box spec flow	k m_{i+u} 0.2	Min. temp. x= 0.967
P6+L10+P6	47.2	0.126	0.691	-0.80	-	0.157	2.587	2.598	0.172	0.160	-	2.532	0.639
P6+L10+P6	37.0	0.125	0.682	-0.79	-	0.147	2.672	2.651	0.174	0.154	-	2.551	0.635
P6+L10+P6	27.3	0.129	0.670	-0.79	-	0.142	2.675	2.621	0.175	0.145	-	2.618	0.630
P6+L20+P6	47.2	0.110	0.691	-0.90	-	0.160	2.625	2.573	0.173	0.174	-	2.421	0.607
P6+L(10-30)+P6	47.6	0.121	0.694	-0.79	-	0.159	2.615	2.492	0.172	0.167	-	2.488	0.610
P6+L(30-10)+P6	48.1	0.129	0.698	-0.66	-	0.158	2.618	2.505	0.176	0.169	-	2.439	0.630
P6+L(30-10)+P6	47.9	0.126	0.697	-0.72	-	0.157	2.618	2.509	0.179	0.172	-	2.398	0.630
P6+L30+P6 No fan	47.7	0.262	0.743	-0.74	-	0.157	2.207	2.139	0.274	0.183	-	1.894	0.657
P6+L40+P6	47.5	0.116	0.696	-0.59	-	0.160	2.619	2.586	0.180	0.180	-	2.331	0.610
P6+L40+P6 No fan	47.1	0.257	0.743	-0.60	-	0.158	2.215	2.160	0.270	0.184	-	1.905	0.655
P6+L40+P6 Fan I+U	48.7	0.135	0.789	-0.59	-	0.152	3.082	2.952	0.123	0.165	-	2.825	0.671
P6+L80+P6	46.5	0.110	0.701	-0.05	-	0.168	2.582	2.521	0.181	0.191	-	2.268	0.621
P6+L80+P6	27.3	0.106	0.684	-0.06	-	0.159	2.649	2.462	0.193	0.193	-	2.183	0.609
P6+L150+P6	48.2	0.120	0.710	-0.05	-	0.158	2.715	2.482	0.186	0.195	-	2.203	0.640

APP.6.3.1 Test window P6, air space 840x1122 mm. Indoor air $\theta_i=1$, outdoor air $\theta_u=0$. θ_{is} and θ_{us} are surface temperatures along the warm and cold surface. The horizontal and vertical gradients are calculated in the centre of the air space. P=plexiglass, G=glass and L=air space. $\Delta\theta$ =difference between indoor-outdoor air temperature ($^{\circ}\text{C}$). Plexi 1 and plexi 2 refer to air space. Hot box refers to mean power through air space+spacer (855x1137 mm).

Case	$\Delta\dot{V}$	θ_{us}	θ_{is}	$\beta_{h, sp1}$	$\beta_{h, sp2}$	$m_{sp1, plexi 1}$	$m_{sp2, plexi 2}$	Spec flow plexi 2	Spec flow plexi 1	Hot-box m_{i+m_u}	Hot-box m_{sp1}	Hot-box m_{sp2}	Hot-box $k_{m_i+m_u=0.2}$	Min. temp. $x=0.967$
P6+L10+P6+L10+G4	47.5	0.080	0.783	-0.80	-0.41	0.141	0.176	1.865	1.812	0.175	0.155	0.188	1.709	1.628
P6+L10+P6+L20+G4	47.4	0.079	0.787	-0.79	-0.37	0.140	0.183	1.862	1.777	0.179	0.160	0.200	1.645	1.579
P6+L10+P6+L40+G4	47.3	0.078	0.788	-0.76	-0.27	0.140	0.185	1.858	1.776	0.177	0.159	0.201	1.653	1.578
P6+L10+P6+L70+G4	47.3	0.074	0.791	-0.78	-0.01	0.141	0.192	1.824	1.769	0.180	0.164	0.217	1.585	1.525
P6+L20+P6+L10+G4	47.3	0.076	0.790	-0.94	-0.33	0.152	0.184	1.836	1.717	0.178	0.174	0.197	1.621	1.553
P6+L20+P6+L20+G4	47.7	0.082	0.793	-0.90	-0.36	0.154	0.190	1.811	1.671	0.182	0.175	0.200	1.603	1.544
P6+L20+P6+L40+G4	47.6	0.077	0.793	-0.88	-0.27	0.154	0.191	1.808	1.682	0.187	0.184	0.212	1.529	1.488
P6+L40+P6+L10+G4	47.3	0.076	0.787	-0.62	-0.37	0.152	0.181	1.865	1.708	0.184	0.180	0.198	1.582	1.532
P6+L40+P6+L10+G4	37.2	0.076	0.780	-0.66	-0.38	0.148	0.172	1.936	1.725	0.184	0.177	0.184	1.626	1.573
P6+L40+P6+L20+G4	47.3	0.074	0.790	-0.61	-0.38	0.150	0.184	1.857	1.721	0.190	0.187	0.214	1.504	1.472
P6+L40+P6+L40+G4	47.4	0.074	0.792	-0.58	-0.25	0.151	0.186	1.858	1.711	0.185	0.183	0.210	1.538	1.491

APP.6.3.2 Test window P6, air space 840x1122 mm. Indoor air $\theta_{i1}=1$, outdoor air $\theta_{u}=0$. θ_{is} and θ_{us} are surface temperatures along the warm and cold surface. The horizontal and vertical gradients are calculated in the centre of the air space. Two air spaces. P=plexi-glass, G=glass and L=air space. $\Delta\dot{V}$ =difference between indoor-outdoor air temperature ($^{\circ}\text{C}$). Plexi 1 and plexi 2 refer to air space. Hot box refers to mean power through air space+spacer (855x1137 mm).

Case	$\Delta\theta$	θ_{us}	θ_{is}	$\beta_{h,sp1}$	$\beta_{h,sp2}$	$m_{sp1,plexi1}$	$m_{sp2-4,plexi2}$	Spec flow plexi1	Spec flow plexi2	Hot-box m_{i-mu}	Hot-box m_{sp1}	Hot-box m_{sp2-4}	Hot-box spec flow	k $m_i+m_u=0.2$	Min. temp. $x=0.967$
P6+L10+P6+L10+P6+L10+G4	47.8	0.064	0.834	-0.86	-0.34	0.132	0.332	1.458	1.333	0.194	0.162	0.372	1.189	1.182	0.796
P6+L10+P6+L10+P6+L20+G4	47.8	0.063	0.836	-0.87	-0.34	0.132	0.347	1.435	1.315	0.207	0.172	0.410	1.100	1.107	0.793
P6+L20+P6+L10+P6+L20+G4	47.6	0.063	0.837	-0.82	-0.35	0.149	0.338	1.428	1.281	0.199	0.188	0.381	1.133	1.130	0.771
P6+L20+P6+L20+P6+L20+G4	47.7	0.060	0.844	-0.83	-0.38	0.144	0.389	1.372	1.178	0.197	0.180	0.414	1.093	1.089	0.765
P6+L20+P6+L20+P6+L10+G4	47.8	0.060	0.842	-0.84	-0.39	0.151	0.374	1.377	1.208	0.200	0.191	0.414	1.087	1.087	0.766
P6+L20+P6+L20+P6+L10+G4	27.0	0.064	0.833	-0.86	-0.42	0.138	0.373	1.563	1.251	0.202	0.189	0.365	1.144	1.147	0.764
P6+L10+P6+L10+P6+L10+G4+L10+G4	46.8	0.054	0.861	-0.82	-0.21	0.127	0.496	1.206	1.088	0.221	0.176	0.620	0.871	0.887	0.830
P6+L10+P6+L10+P6+L10+G4+L10+G4	27.2	0.054	0.850	-0.82	-0.22	0.119	0.440	1.357	1.158	0.196	0.156	0.489	1.041	1.037	0.821
P6+L20+P6+L20+P6+L20+G4+L10+G4	46.6	0.049	0.871	-0.86	-0.27	0.144	0.563	1.171	0.973	0.203	0.193	0.622	0.877	0.879	0.799

APP.6.3.3 Test window P6, air space 840x1122 mm. Indoor air $\theta_i=1$, outdoor air $\theta_u=0$. θ_{is} and θ_{us} are surface temperatures along the warm and cold surface. The horizontal and vertical gradients are calculated in the centre of the air space. Three, four air spaces. P=plexiglass, G=glass and L=air space. $\Delta\theta$ =difference between indoor-outdoor air temperature ($^{\circ}\text{C}$). Plexi 1 and plexi 2 refer to air space. Hot box refers to mean power through air space+spacer (855x1137 mm).

Case	$\Delta\dot{V}$	θ_{us}	θ_{is}	θ_{tf}	$m_{sp1} \cdot m_{U_{sp1}}$	m_{sp1}	k $m_i + m_{U_{sp1}} = 0.2$	Min. temp. $x=1.00$	Min. temp. $x=0.964$			
D4-6	22.2	0.147	0.557	0.743	0.114	0.172	3.134	0.506	0.542			
D4-6	32.5	0.157	0.567	0.750	0.111	0.169	3.155	0.526	0.562			
D4-12	22.4	0.127	0.607	0.750	0.172	0.172	2.794	0.466	0.563			
D4-12	32.5	0.137	0.627	0.760	0.179	0.169	2.761	0.476	0.575			
D4-12	47.0	0.137	0.643	0.770	0.187	0.164	2.717	0.488	0.593			
D4-15	22.0	0.127	0.623	0.757	0.206	0.171	2.707	0.509	0.551			
D4-15	32.3	0.127	0.640	0.763	0.216	0.168	2.665	0.526	0.557			
D4-15	47.1	0.130	0.653	0.773	0.224	0.162	2.646	0.536	0.581			
D4-6, SF ₆	22.4	0.127	0.623	0.747	0.200	0.166	2.753	0.478	0.539			
D4-6, SF ₆	32.5	0.133	0.630	0.757	0.202	0.163	2.768	0.495	0.548			
D4-6, SF ₆	47.2	0.140	0.637	0.763	0.201	0.157	2.806	0.507	0.579			
D4-12, SF ₆	22.1	0.127	0.610	0.753	0.216	0.166	2.810	0.453	0.523			
D4-12, SF ₆	32.5	0.133	0.620	0.763	0.186	0.163	2.821	0.483	0.539			
D4-12, SF ₆	46.9	0.140	0.633	0.770	0.190	0.158	2.828	0.499	0.570			

APP.6.4.1

Test window P7, air space 834x1116 mm. Indoor air $\theta_i=1$, outdoor air $\theta_u=0$.

θ_{us} are surface temperatures along the warm and cold surface. D4-12, 4 mm double panes, 12 mm air space. T=triple glazed, LF=low emissivity layer (kappaenergi), (-)=layer turned towards cold side. SF₆ and Ar= gases in the sealed unit. $\Delta\dot{V}$ =measured difference between indoor-outdoor air temperature ($^{\circ}\text{C}$), θ_{if} =mean temperature of frame on the inside, m_{flm} =mean resistance of space (s) measured with heat flow sensor. m_{sp1} and k have been measured with the hot box through the whole sealed unit (855x1137 mm).

Case	$\Delta\theta$	θ_{us}	θ_{ts}	θ_{tf}	m_{f1m} $sp1$	m_i+m_u	m_{sp1}	k $m_i+m_u=$ 0.2	Min. temp. $x=1.00$	Min. temp. $x=$ 0.964				
D4-15, SF6	22.0	0.123	0.620	0.753	0.199	0.166	0.155	2.738	0.456	0.529				
D4-15, SF6	32.1	0.137	0.633	0.763	0.197	0.163	0.154	2.749	0.477	0.545				
D4-15, SF6	46.8	0.140	0.647	0.773	0.200	0.162	0.156	2.730	0.480	0.576				
D4-12, SF6 No fans	22.2	0.263	0.677	0.793	0.214	0.242	0.160	2.700	0.541	0.587				
D4-12, SF6 No fans	32.6	0.280	0.690	0.803	0.223	0.245	0.159	2.710	0.552	0.601				
D4-12, SF6 No fans	46.8	0.293	0.700	0.813	0.247	0.248	0.159	2.707	0.575	0.628				
D4-6, Ar	22.4	0.140	0.603	0.747	0.169	0.170	0.137	2.880	0.499	0.569				
D4-6, Ar	32.4	0.140	0.617	0.757	0.177	0.167	0.142	2.840	0.515	0.589				
D4-6, Ar	46.9	0.143	0.633	0.767	0.184	0.163	0.148	2.795	0.531	0.614				
D4-12, Ar	22.1	0.123	0.636	0.760	0.214	0.175	0.174	2.605	0.582	0.628				
D4-12, Ar	32.5	0.123	0.657	0.767	0.229	0.167	0.180	2.565	0.591	0.646				
D4-12, Ar	47.1	0.123	0.673	0.773	0.245	0.162	0.187	2.518	0.609	0.669				
D4-15, Ar	22.1	0.120	0.643	0.760	0.223	0.174	0.181	2.558	0.515	0.579				
D4-15, Ar	32.5	0.120	0.657	0.763	0.240	0.168	0.185	2.531	0.521	0.586				
D4-15, Ar	47.2	0.127	0.673	0.777	0.237	0.163	0.187	2.518	0.518	0.605				

APP.6.4.1 (cont)

Case	$\Delta\tilde{w}$	θ_{us}	θ_{is}	θ_{if}	$m_{f m_{sp1+2}}$	m_i+m_u	m_{sp1+2}	$k_{m_i+m_u=0.2}$	Min. temp. $x=1.00$	Min. temp. $x=0.964$		
D4-12, LE	22.2	0.097	0.713	0.763	0.340	0.174	0.268	2.091	0.559	0.661		
D4-12, LE	32.7	0.103	0.720	0.773	0.329	0.172	0.264	2.109	0.570	0.659		
D4-12, LE	47.1	0.106	0.723	0.780	0.329	0.166	0.262	2.119	0.569	0.652		
D4-12, LE, Ar	22.2	0.077	0.767	0.770	0.506	0.172	0.368	1.729	0.600	0.673		
D4-12, LE, Ar	32.4	0.083	0.777	0.780	0.523	0.163	0.360	1.754	0.600	0.671		
D4-12, LE, Ar	47.2	0.087	0.777	0.787	0.505	0.158	0.346	1.797	0.605	0.666		
T4-12	22.1	0.087	0.723	0.787	0.364	0.171	0.283	2.007	0.575	0.671		
T4-12	32.5	0.090	0.740	0.797	0.379	0.170	0.298	1.949	0.583	0.686		
T4-12	46.9	0.093	0.750	0.803	0.392	0.170	0.312	1.898	0.592	0.694		
T4-12, Ar	22.5	0.083	0.747	0.793	0.423	0.170	0.321	1.866	0.611	0.692		
T4-12, Ar	32.5	0.083	0.760	0.800	0.439	0.166	0.330	1.834	0.619	0.702		
T4-12, Ar	47.4	0.083	0.773	0.807	0.467	0.161	0.344	1.787	0.627	0.710		
T4-12, Ar, LE	22.3	0.063	0.813	0.790	0.712	0.170	0.492	1.414	0.599	0.741		
T4-12, Ar, LE	32.6	0.063	0.820	0.797	0.727	0.161	0.493	1.412	0.603	0.743		
T4-12, Ar, LE	47.1	0.063	0.830	0.807	0.757	0.163	0.520	1.361	0.605	0.742		

Case	$\Delta\sigma$	θ_{us}	θ_{is}	θ_{if}	m_{f1m} $sp1+2$	m_i+m_u	m_{sp1+2}	k $m_i+m_u=$ $x=1.00$	Min. temp. $x=1.00$	Min. temp. $x=$ 0.964				
T4-12, LE(-), Ar	22.4	0.060	0.810	0.797	0.701	0.165	0.482	1.434	0.628	0.736				
T4-12, LE(-), Ar	32.7	0.063	0.820	0.807	0.698	0.171	0.514	1.372	0.635	0.740				
T4-12, LE(-), Ar	48.1	0.060	0.823	0.810	0.708	0.158	0.496	1.406	0.633	0.738				
D4-12, LE, Ar, rebate 5 mm	22.2	0.067	0.767	0.763	0.505	0.164	0.371	1.722	0.597	0.668				
D4-12, LE, Ar, rebate 5 mm	32.8	0.087	0.780	0.780	0.509	0.176	0.388	1.674	0.606	0.669				
D4-12, LE, Ar, rebate 5 mm	47.0	0.080	0.777	0.783	0.482	0.158	0.352	1.780	0.599	0.661				
D4-12, LE(-)	32.9	0.103	0.723	0.777	0.339	0.172	0.268	2.091	0.556	0.646				
T4-12, LE, Ar, rebate 5 mm	32.7	0.060	0.823	0.800	0.746	0.166	0.516	1.369	0.650	0.739				
D4-12 h=1117	22.1	0.131	0.613	0.737	0.169	0.169	0.146	2.810	0.499	0.581				
D4-12 h=1117	32.3	0.130	0.627	0.745	0.177	0.166	0.154	2.751	0.504	0.591				
D4-12 h=1117	32.4	0.135	0.627	0.745	0.173	0.167	0.152	2.759	0.506	0.592				
D4-12 h=1117	47.0	0.133	0.643	0.757	0.182	0.167	0.163	2.681	0.519	0.603				
T4-12 h=1117	22.3	0.090	0.727	0.763	0.355	0.174	0.287	1.991	0.584	0.673				
T4-12 h=1117	32.7	0.090	0.740	0.770	0.365	0.166	0.293	1.967	0.591	0.682				
T4-12 h=1117	47.0	0.090	0.750	0.773	0.372	0.163	0.303	1.932	0.593	0.686				

APP.6.4.1 (cont)

	δ_u (°C)	Fan	d_1 (mm)	Dust strip	d_2 (mm)	θ_{ig}	θ_{ug}	θ_{ic}	θ_{uc}	θ_{if}	θ_{uf}	$\frac{m_i+m_u}{m^2K/W}$	$\frac{m_{ss}}{m^2K/W}$	$\frac{k}{m_i+m_u=0.2}$
SP-window W-IF1 1.2x1.2m	0	II	2.4	-	2.8	0.752	0.096	0.821	0.063	0.779	0.037	0.184	0.407	1.65
	-10	II	2.4	-	2.8	0.756	0.090	0.830	0.061	0.780	0.036	0.177	0.406	1.65
	-25	II	2.4	-	2.8	0.755	0.087	0.835	0.061	0.782	0.036	0.173	0.404	1.66
	-10	I	2.4	-	2.8	0.756	0.110	0.839	0.096	0.797	0.052	0.196	0.411	1.64
	-10	Ø	2.4	-	2.8	0.770	0.191	0.855	0.189	0.822	0.141	0.256	0.418	1.62
	-10	II	2.4	Taped	2.8	0.754	0.095	0.838	0.070	0.791	0.038	0.189	0.423	1.61
	-10	Ø	2.4	Taped	2.8	0.787	0.220	0.868	0.208	0.838	0.141	0.298	0.460	1.52
	-10	II	2.4	Dust strip	2.8	0.751	0.095	0.837	0.068	0.795	0.038	0.190	0.422	1.61
	-10	Ø	2.4	Dust strip	2.8	0.785	0.213	0.866	0.198	0.838	0.137	0.290	0.454	1.53
	-10	II	5.0	Dust strip	2.8	0.752	0.100	0.839	0.071	0.794	0.041	0.193	0.419	1.62
	-10	Ø	5.0	Dust strip	2.8	0.785	0.222	0.868	0.206	0.837	0.141	0.310	0.456	1.53
	-10	II	5.0	-	2.8	0.743	0.090	0.828	0.057	0.784	0.036	0.182	0.400	1.67
	-10	Ø	5.0	-	2.8	0.765	0.193	0.852	0.168	0.818	0.134	0.254	0.408	1.65
	0	II	5.0	-	2.8	0.733	0.090	0.818	0.049	0.775	0.028	0.187	0.396	1.68
	-10	II	7.0	-	2.8	0.740	0.087	0.827	0.055	0.781	0.037	0.183	0.399	1.67

APP.6.6.1 Coupled window of variable gap width. Indoor air $\theta_{i1}=1$, outdoor air $\theta_u=0$. θ_{ig} , θ_{ic} and θ_{if} are mean temperatures on the inner surface of glass, casement and frame respectively. θ_{ug} , θ_{uc} and θ_{uf} are corresponding temperatures for the outer surface. $\theta_{o1}=22.5$ °C, for d_1 and d_2 see FIG.6.6.1. Fan Ø, I and II denote air velocities of 0, 0.5 and 1.0 m/s respectively in the cold room. m_{ss} =inner to outer surface resistance.

	ϑ_u (°C)	Fan	d_1 (mm)	Dust strip	d_2 (mm)	θ_{ig}	θ_{ug}	θ_{ic}	θ_{uc}	θ_{if}	θ_{uf}	m_i+m_u m^2K/W	m_{ss} m^2K/W	k $m_1+m_u=$ 0.2
SP-window W-IF1 1.2x1.2 m	-10	Ø	7.0	-	2.8	0.757	0.172	0.849	0.154	0.813	0.123	0.235	0.396	1.68
	-10	II	7.0		2.8	0.749	0.093	0.833	0.063	0.786	0.032	0.189	0.417	1.62
	-10	Ø	7.0		2.8	0.784	0.219	0.867	0.205	0.838	0.139	0.296	0.457	1.52
	-10	II	7.0	-	5.8	0.736	0.084	0.816	0.057	0.766	0.031	0.175	0.376	1.74
	-10	Ø	7.0	-	5.8	0.751	0.165	0.846	0.165	0.808	0.121	0.228	0.386	1.71
	-10	II	7.0	-	9.4	0.737	0.083	0.811	0.057	0.767	0.031	0.174	0.375	1.74
	-10	Ø	7.0	-	9.4	0.749	0.163	0.843	0.171	0.811	0.121	0.225	0.381	1.72
	-25	II	7.0	-	9.4	0.744	0.076	0.818	0.054	0.769	0.029	0.166	0.376	1.74
	0	II	7.0	-	9.4	0.727	0.084	0.802	0.050	0.757	0.027	0.177	0.365	1.77

	Fan II, $\vartheta_U \sim -10$ °C				Fan Ø, $\vartheta_U \sim -10$ °C			
	Warm		Cold		Warm		Cold	
	°C	T	°C	T	°C	T	°C	T
Frame								
t	17.87	0.854	-9.37	0.037	20.00	0.916	-2.48	0.237
s	16.43	0.811	-9.44	0.035	18.54	0.871	-6.28	0.123
s h	12.18	0.683	-9.53	0.032	17.87	0.851	-6.22	0.125
b	15.15	0.772	-9.30	0.039	15.62	0.783	-7.73	0.079
Casement								
t	19.14	0.892	-7.88	0.082	19.44	0.899	0.75	0.335
s	17.53	0.844	-8.11	0.075	17.92	0.853	-3.73	0.200
s h	16.16	0.803	-8.99	0.048	14.90	0.762	-4.58	0.174
b	15.5	0.783	-9.35	0.037	15.29	0.773	-8.84	0.046
Glass								
x=51	16.77	0.821	-6.55	0.121	17.56	0.842	-1.35	0.271
x=276	14.75	0.760	-7.88	0.082	15.60	0.783	-3.71	0.200
x=501	14.19	0.744	-7.70	0.087	14.94	0.763	-4.27	0.183
x=726	14.10	0.741	-7.54	0.092	14.50	0.750	-4.60	0.173
x=951	13.45	0.721	-8.16	0.073	13.54	0.721	-5.87	0.135
Air	22.74	1.000	-10.60	0.000	22.80	1.000	-10.35	0.000
Rainwater channel								
t	-6.03	0.137	-	-	6.02	0.494	-	-
s	-5.71	0.147	-	-	0.33	0.322	-	-
s h	-8.29	0.069	-	-	-1.17	0.277	-	-
b	-5.26	0.160	-	-	-6.16	0.126	-	-
b	2.83	0.403	-	-	2.71	0.394	-	-

APP.6.6.2 Surface temperature on frame, casement and glass on the warm and cold side. Also air temperatures in the hot box, cold room and rainwater channels in the window. θ =nondimensional temperature, $\theta_i=1$, $\theta_u=0$, $d_1=2.4$ mm and $d_2=2.8$ mm according to FIG.6.6.1. t, b and s = top rail, bottom rail and stile respectively. h = hinge.

	Δt_u (°C)	$\frac{\theta}{z_1/b=0.500}$	$\frac{\theta}{z_1/b=0.822}$	$\frac{\theta}{z_1/b=0.500}$	$\frac{\theta}{z_1/b=0.822}$	$\delta_i(x)$	Δt_i (°C)						
Window 1 $x_1/L=-0.034$						0.980							
0.133		0.196		0.690	0.678	1.000							
0.500	3.75	0.153		0.648		0.981	24.28						
0.867		0.136	0.127	0.610	0.613	0.961							
1.034						0.908							
Window 1 -0.034						0.981							
Taped 0.133		0.197		0.689	0.681	0.999							
0.500	5.78	0.149		0.648		0.983	24.01						
0.867		0.152	0.130	0.614	0.617	0.964							
1.034						0.892							
Window 2 -0.055						1.006							
0.146		0.210		0.706	0.691	1.007							
0.500	6.87	0.182	0.170	0.657		0.984	29.51						
0.854		0.134		0.620	0.623	0.961							
1.055						0.878							

APP.6.7.1 Temperatures on the outside and inside of the glass and air temperatures on the outside and inside.

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